PRODUCT ENGINEERING

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Production's Demands on Design

ENGINEERING departments throughout the machinery industry are now getting drastic tests of the designs they developed during the quieter depression years. Their products, of course, were given searching tests before they were offered to customers, but even the most thorough engineering tests seldom give quite the same result as does the abuse handed out by a careless or ignorant user.

Although automobile proving ground testing has been developed to a remarkable degree, it is not always possible to subject a car to all of the grief a poor driver will give it. One company has gone so far as to put on its test staff the worst driver it could find but users are managing to break parts that withstood his efforts.

As weaknesses in a product are uncovered by field failures the design department must step in quickly and make the substitution of materials or the modification in structure necessary to cure the trouble. At the present time this condition is acute in the plants making production machinery. It is more serious, of course, with the equipment that is not new and has had a chance to deteriorate through wear and hard treatment. But it also exists where some of the newer units are being run to the limit to keep up with production demands.

Production is going ahead full steam in most branches of the metal-working industry, and that means peak demand on the equipment. Under such conditions operators of machines are more likely to force their machines than they are to worry about keeping things finely adjusted. For example, turret locking rings on one well known machine are clamped by a cast iron lever amply strong for the usual requirements so long as the parts are in adjustment. But men operating these machines in high production shops are neglecting adjustments and levers are causing trouble. Under the circumstances it is easier to substitute a cast steel lever than to fight about having the proper adjustments made when they should be.

IT MAY NOT be the best engineering to handle the job this way, but there come times, as every engineer knows, when conditions rather than principles take control. Regardless of the cost of making changes in detail before a model run is completed these changes must be made when the demand comes. The best the engineers can do is to try to anticipate such troubles and be prepared for them with a quick remedy when they crop up.

As production demands step up still more and the loads on individual units increase, we can expect more demands on design for emergency changes. The engineers will meet them and find the right answers. That's what engineers are for.

Every Finishing Specification Is a Special Problem

GUSTAVE KLINKENSTEIN
Vice-President and Technical Director
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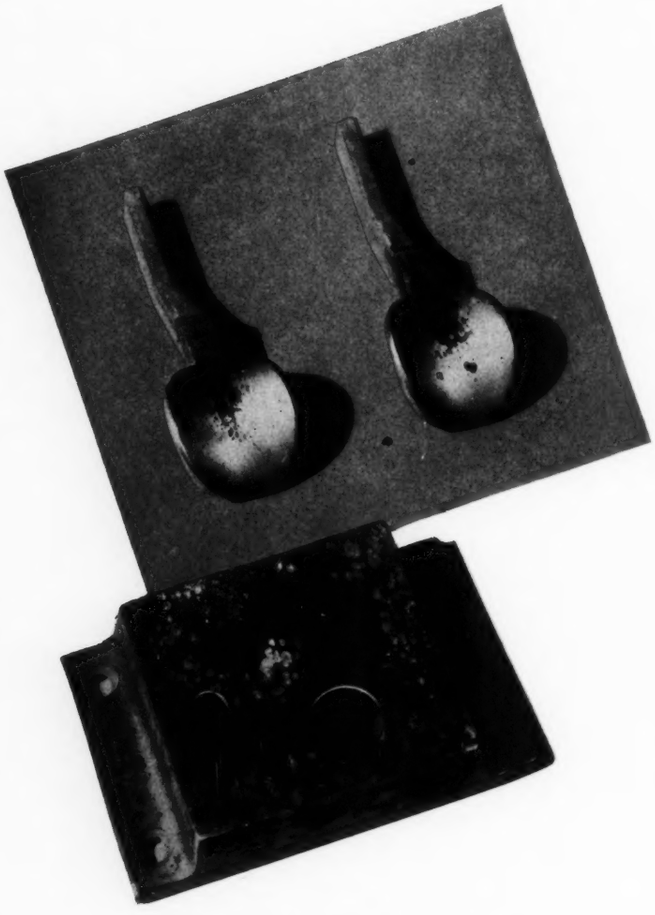
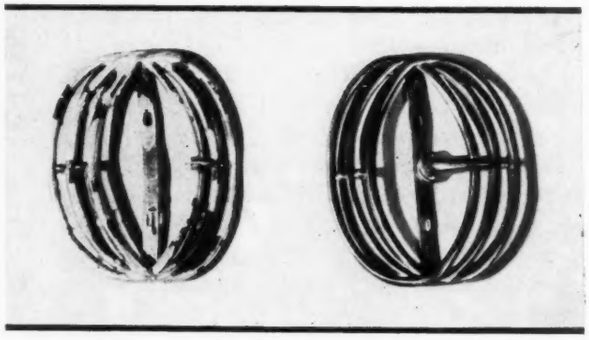


These brass-plated malleable-iron lock parts are badly stain-spotted because the lacquer is being attacked by the traces of chemicals used in brass plating. Special lacquers for brass plating reduce to a minimum the rejects from this cause



Oxidized silver belt buckle with lacquer peeling off and taking the silver sulphide film along with it. The same buckle finished with correct lacquer is also shown

Resistance to moisture, alkalies, acids and oils are only a few of more than 45 qualities possessed by finishes, any specific combination of which governs the formulation for a particular application

"IN America, I was surprised to find that even the most costly lacquered articles, made of brass, bronze, silver and other metals, frequently revealed faulty films, showing iridescence, bloom, flaking and other defects."

This is severe criticism, but it comes from one who is well qualified to speak; namely, Dr. Fritz Zimmer, a leading German authority on finishes.

And Dr. Zimmer puts his finger squarely on one of the important reasons for inferior product finishing in this country by pointing out that Americans prefer to meet every problem by a general solution.

There is no doubt that we Americans are thoroughly imbued with the advantages of standardization of products, simplification of lines, and elimination of special services. Much of the success of our leading manufacturers in almost all lines can be attributed to the fact that they have adhered to these principles of production and have cut out off-standard operations whereever possible. But there are some places where they simply will not work, and the specification and application of finishes is one of them.

It is unfortunate that a few standard finishes and finishing schedules cannot be applied universally. It would be much easier for both user and maker if a limited number of finishes could be turned out by mass production methods and used successfully by manufacturers of many different kinds of products. But in the present state of the art of making finishes, this cannot be done. The manufacturer of finishes, if he is to serve his customers to their best advantage, must resign himself to all the drawbacks of formulating and producing innumerable special products and he must spend much time in convincing his customers that to use a lacquer, or lacquer enamel, or thinner for any purpose other than that for which it is intended may be disastrous.

It is, for example, rather common practice for a manufacturer or metal finisher to use the same clear

lacquer for silver, brass, bronze, and sometimes other metals, with apparently satisfactory results. This may be quite true as far as one can judge by looking at the finished products in the factory, but let us look a little further into this matter.

As an example of the problems encountered in selecting finishing materials, one of the requirements of a good lacquer for silver is that it shall be so clear as to be invisible when applied to the surface of the metal. Such lacquer naturally commands a higher price. It will likewise provide a nice looking finish for brass, bronze, or copper, but why go to the expense of using a perfectly clear higher priced lacquer for these latter metals? On the other hand, if a lacquer intended for bronze is used on silver, it is likely to color the metal slightly, thereby decreasing the quality appearance of the product.

It must also be considered that neither the metal nor the lacquer film is inert chemically. If wrongly selected, the film may react with its metal base or with any of the various metallic chemical compounds formed in the process of "tarnishing" or "oxidation." In either case, the protecting film will be damaged and the product will rapidly deteriorate in appearance.

Products of solid or plated silver are rarely used out-of-doors, whereas products of other metals may be exposed to the weather. An entirely different type of film is needed for outdoor exposure in order to withstand the destructive actions of the elements.

That which may not be detrimental in one application may be a serious defect in another. When a piece of rubber comes in contact with lacquered silverware — the silver surface rapidly tarnishes because the sulphur in the rubber penetrates through the lacquer and reacts with the silver, forming silver sulphide. This occurs because silver lacquer is not intended to resist the action of sulphur. But the lacquer that covers the brass ferrule holding the eraser at the end of a common pencil must be formulated with this very point in mind, to prevent the pencil from quickly becoming unsalable.

Cases are also frequently encountered where a lacquer that is good for solid brass is also being used for brass-plated goods. This is a natural thing to do because one is apparently dealing with surfaces of the same chemical composition. But this is not true. No matter how carefully articles are washed after plating, they are bound to carry traces of the plating chemicals in the pores of the applied metals, and unless the lacquer used is designed to withstand the action of these chemicals, the goods will in time become discolored with dark spots.

Finishing defects, such as those mentioned, may not become evident until some time after the products have been shipped. The manufacturer may know nothing about them until complaints and rejects begin to pour in. Heavy losses may be sustained in this way, so it is poor policy to take chances with the wrong lacquer.

Finishes Must All Be Special

In general, a special lacquer must be formulated to suit the chemical characteristics of each metal and alloy and their "oxidation" products. Sometimes the chemical problem involved is an extremely difficult one. Aluminum, zinc, cadmium, and some other metals tend to form metallic "soaps" with common lacquer bases, and only recently has it become possible for manufacturers to secure lacquers that can be applied directly to these metals without some kind of pre-treatment. Chromium offers special difficulties. No known air-drying lacquer or lacquer enamel can be applied directly to a chromium-plated product with satisfactory results. The best practice is first to bake on a special clear primer, after which almost any kind of finish can be used.

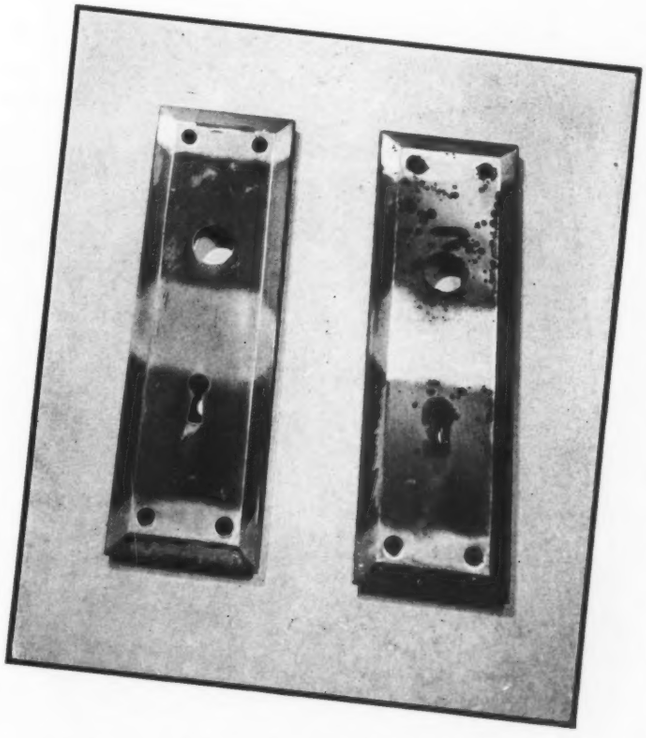
Naturally, for materials other than the metals, such as wood of various kinds, leather, paper, textiles and plastics, still other types of lacquer are needed.

But the chemical characteristics of the material to be finished provide only a few of the factors entering into the problem of finish formulation. The physical character of the material must also be considered. Is the surface smooth or rough; compact or porous; flat, rounded, or irregular?

Service requirements vary widely. Some products, like clocks and picture frames, are by their nature carefully protected, but others are subjected to the roughest of treatment and their protective finish may be required to be resistant to such destructive agencies as marring, dampness, extremes of heat or cold, acids and alkalies, fruit juices and alcoholic beverages, household cleansers and chemicals, grease and oil, lighted cigarettes and matches or innumerable other conditions.

Some finishes must be tough and flexible; others hard and durable; others soft and easily sanded. Some must dry naturally with a high, medium, or low gloss, or with no gloss at all; some must be capable of developing a brilliant polish under rubbing and buffing. Some must dry rapidly, others must remain sticky for a definite length of time, others must be suitable for baking.

Color introduces additional complications. Not only have the various pigments chemical and physical properties that have to be taken into consideration, but it



Examples of door plates finished in oxidized copper which have been coated with an ineffective lacquer. One shows film breaking down at screw holes and along edges as a result of exposure to weather, the other has a film which is unable to withstand the chemical action of copper sulphide

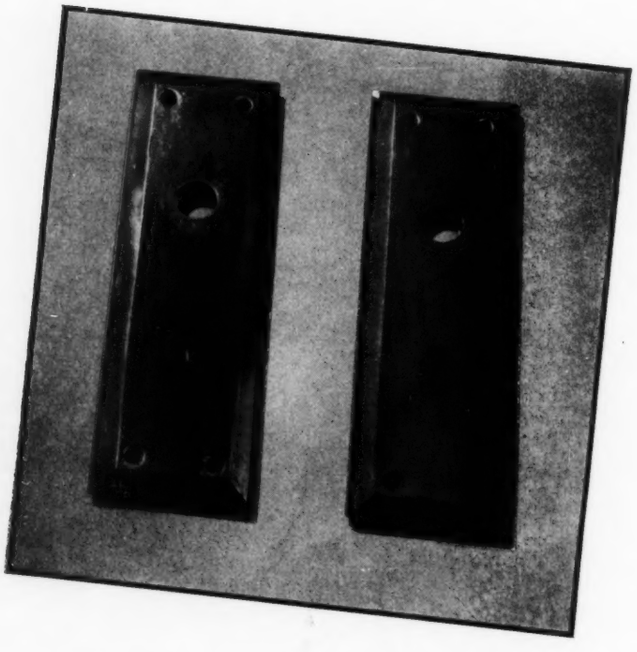
must be possible to supply almost every pigmented finish in any desired shade. Finish manufacturers have standard color cards, but they are of little use; most manufacturers want "something different" for their products.

Progress Causes Complications

The finishing requirements that have been cited by no means exhaust the possibilities, but they are enough to show why the modern manufacturer of finishes for industrial products has had to abandon all ideas of standardized, mass production and is compelled to treat practically every product as a special case.

Twenty-five years ago, conditions in the finishing business were different. At that time, the manufacturer of products had at his command only a limited number of finishes. These consisted of paints, varnishes, enamels, and a few clear lacquers, all of which had been manufactured for years with but minor changes in their formulas. Every experienced finisher knew all he needed to know about each one of these and he was capable of using each to the best advantage.

Chemical science has made available thousands of raw materials suitable for finish manufacture — lacquer bases, solid and liquid plasticizers, natural and synthetic resins, pigments, solvents, and thinners — and new ones are being added every day. Each of these materials, either alone or in combination with others, imparts certain properties to the finish, and, by proper selection and processing, finishes with qualities undreamed of twenty-five years ago can now be procured. But it has become impossible for anyone who has not devoted years to specializing in the subject to know the details of these new finishes or to know what can be expected of each. Therefore, it is necessary for the manufacturer to rely on experts, if he is to get the most satisfactory finish.



Illustrating how lacquer fails along the edges, around screw holes and generally over the surface when brass door plates finished with lacquer intended for interior use are installed on outside of door. The film of a lacquer for exterior use would be intact after the same exposure

The Finishing Engineer Appears

This state of affairs has brought a new figure into the industrial world — the finishing engineer, who may be an independent consultant or, more usually, a representative of a finishing manufacturer. In serving the product manufacturer, the finishing engineer makes a survey which includes:

1. The raw material of the product.
2. Its process of manufacture — whether it is stamped, pressed, molded, die-cast, machined, electro-plated, engraved or other process.
3. The character of the surface of the product when the finishing stage is reached — whether dirty, clean, scaly, oily, or greasy.
4. The character of the finishing facilities available — which may vary from a finishing department, with air-conditioned, dust-free rooms with modern spraying and dipping equipment, baking ovens, and expert finishers, to a mere enclosure manned by common labor.
5. The service requirements of the product.
6. Marketing details — such as quality, price, method of shipment, distribution and display, and the sales appeal of color.

After his study the finishing engineer has the alternative to supply a finish that will exactly suit conditions as he finds them, or he can recommend changes in those conditions in order to reduce the finishing costs or to secure a more salable product by virtue of a better finish.

For example, the finishing engineer must frequently recommend one of the three following courses to a manufacturer: (*a*) use a finish adapted to existing crude facilities, (*b*) install thoroughly modern finishing equipment, (*c*) have the finishing done by outside specialists.

Another common problem is this: Should the manufacturer use a finish that can be applied to the product just as it comes from the final processing machine, or should a complete cleaning, degreasing, and rust-proofing system be installed

Frequently the finishing engineer has to criticize a design. "Round off this sharp corner and fill up this depression, or else look for trouble in the finishing room," he may say. Or he may recommend that certain products be finished before stamping, forming, or machining.

The question of color is often of major importance. Shall black, white, gray, or cream-color be adopted as standard or shall a gayer red, blue, yellow, or other shade be used. If the latter, will a single color suffice for all users, or will it be necessary to supply a range of colors in order to match decorative schemes?

Not until these and many other questions have been answered and an agreement as to detail has been reached with the product manufacturer, is the finishing engineer ready to recommend a definite finishing schedule and prepare specifications for the specific finishes to be supplied. Nor is his work done when production of the product is fairly under way and everything is apparently in satisfactory shape. He must keep in constant touch with the product manufacturer and be ready at any time to furnish a revised formula in order to meet changes in conditions, process, or requirements.

From what has been said it should be clear that the finish of a product is not a trivial matter to be taken up at the last moment after every other detail of design and production has been settled.

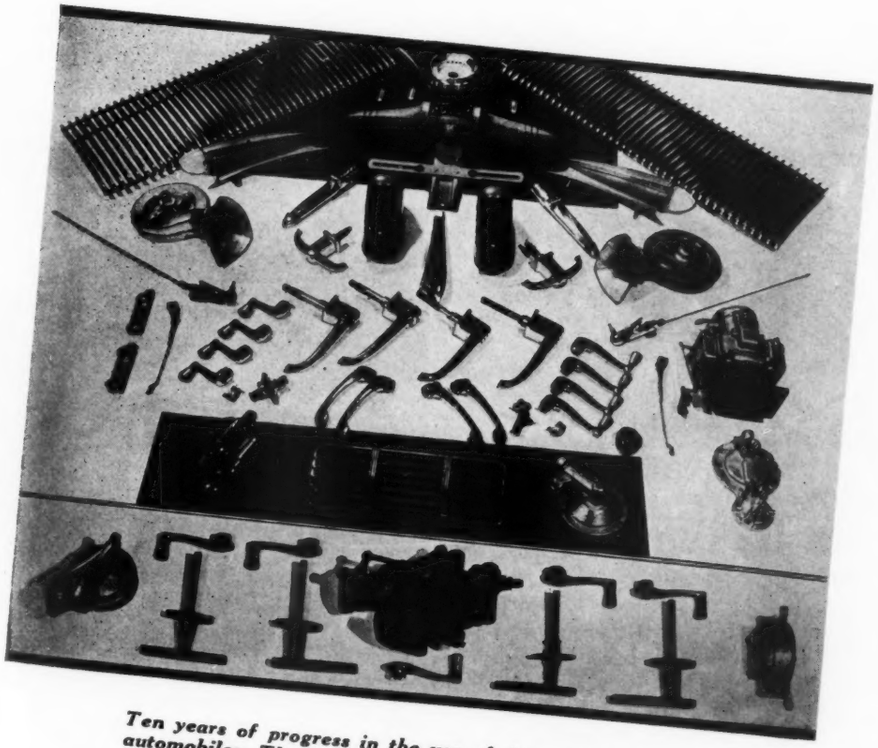
PRODUCT DEVELOPMENTS

Current Buick Uses Fifty Die-Castings

● Whereas ten years ago Buick was using 12 die-castings as shown in the lower portion of the accompanying illustration, the 1937 model is now using 50 die-castings. In the upper part of the illustration are shown the parts that are used on the inside and outside of the car.

Inside the car the steering wheel hub and spacers are die cast, as are the radio grill, robe cord door handles, sun visor, and window regulators. Die castings used under the hood include the carburetor, clutch release bearing, fuel pump, and oil pump cover. On the exterior of the car are die-cast radiator grills and ornaments, windshield wiper housing and arm, license plate lamp, fender and tail lamps, door handles and trunk hinges.

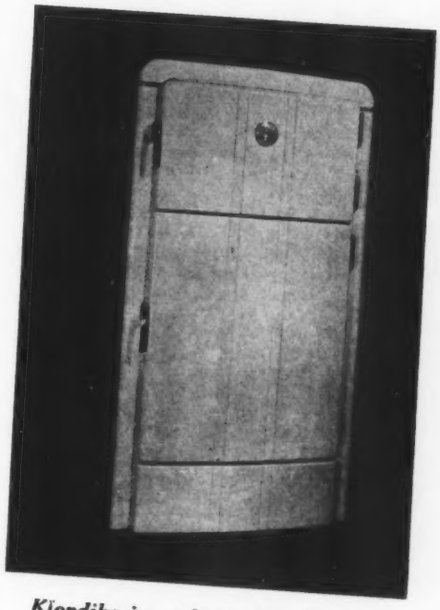
As a comparison of 9½ lb. of zinc die castings used on the 1927 Buick, 51 lb. of die castings are used on the 1937 Buick. It is interesting to note that the various applications of die castings to the car ten years ago are still being continued on the new model.



Ten years of progress in the use of die castings in automobiles. The 12 pieces in the lower section of the illustration were used in 1927 Buicks. Above them are the 50 die cast parts in the latest model

● *Modern Materials in Refrigerators*

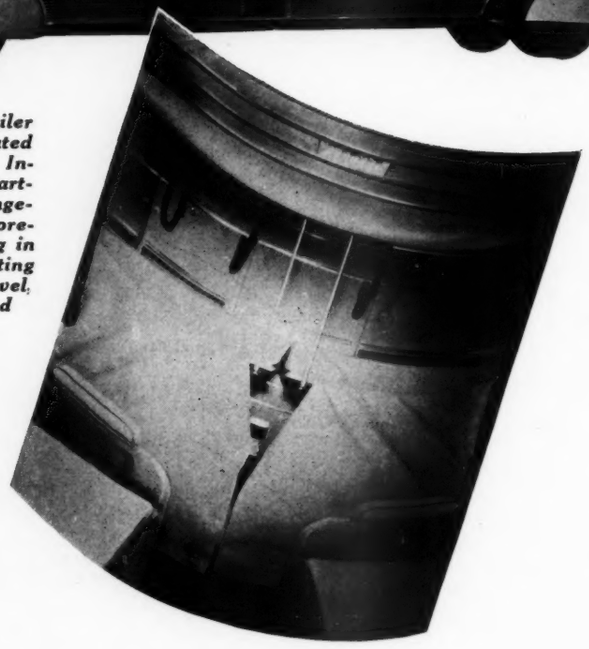
In keeping with current designs of electric refrigerators, the new ice refrigerator made by the Maine Manufacturing Company is fabricated of modern materials in place of golden oak formerly used. The case is finished inside and outside with baked enamel, the floor of the food chamber being of porcelain enamel. For insulation, balsam wool fiber, 3 in. thick, is used in waterproof packing sealed with asphalt. Trim, locks, handles and hinges, made of brass, are triple-chromium-plated. In contrast to the light colored case, the base is finished in black. The cover for the waste water pan compartment is hinged on extension sliding bars and swings upwards.



Klondike ice refrigerator designed by Robert Heller, is finished in enamel baked on steel with chromium trim and hardware

● *1,000 Hp. by V-Belts*

To reduce weight in the propeller shaft drive of the new speed boat, Miami Queen, the use of V-belts and aluminum pulleys saves 75 per cent in weight over a gear drive. Two 12-cylinder Liberty engines, each rated 450 hp. at 2,000 r.p.m., are used in the radically new design of hull with which it is expected to exceed the speed of 124 mi. per hr., the present record. Each engine is connected to a propeller shaft by 30 Gates V-belts running on aluminum sheaves. In tests with engines turning up to 2,400 r.p.m. and the propeller shafts rotating at more than 6,000 r.p.m., the belt speed was 13,000 ft. per min. Although such a belt speed is much greater than used in any commercial installations, both the belts and sheaves have withstood this high speed for more than a total of 5 hr. of test runs.



Light-weight sleeper-trailer of stainless steel fabricated by "shotweld" process. Interior showing compartments. Showing arrangement for day travel in foreground and for sleeping in the background. By locating lower berth near floor level, full height is obtained

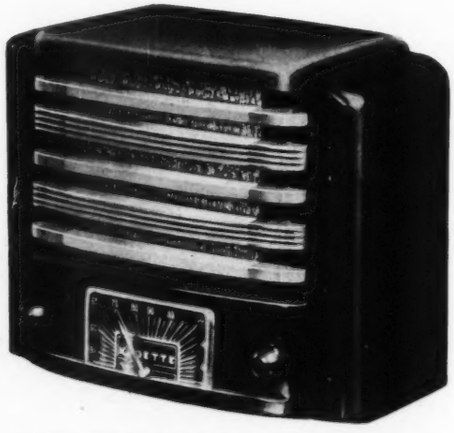
Sleeper-Trailer Designed Like High-Speed Train

● First of its type ever to be constructed, a light-weight, stainless steel sleeper-bus has been designed and manufactured by the Edward G. Budd Manufacturing Company for use in the Syrian desert between Bagdad and Damascus. This 14-passenger trailer is air conditioned and follows closely the design of railway equipment built by the Budd Company of light-weight stainless steel by the "shotweld" process. It was so designed to reduce weight and for a top speed of 65 mi. per hr., under bad road conditions. It is expected that the 600-mi. desert run will be accomplished in 15 hours, in comparison to the 24-hr. schedule now in effect between the two cities.

A 150-hp. tractor powered by a diesel engine pulls the trailer. Especially constructed 10-ply rubber tires carrying 25 to 30 lb. of air are used for both the tractor and the trailer. The overall length is 57 ft. 6 in., and that of the trailer is 36 ft. 8 in., with a height of 8 ft., 7 in. when loaded. The total weight of the bus is 28,000 lb. Whereas in other vehicles used for desert service, it is customary to merely shut off the power and coast to a stop, this bus has air-brakes for quick stopping.

At the point of attachment to the spring hangers which are made of Cromansil, and the 72 in. springs, the floor beams are increased in size to meet the loading conditions. The floor beams are 6-in. channel shaped sections between the front and rear cut-outs. A strong alloy cross member is built into the body at the lower edge of the front cut-out for reinforcement to prevent this end from striking the ground. Also, where the structure is cut out, such as around the rear axle and front end, reinforcements are added to reduce localized stresses. The roof is of standard design except that the rear end is pointed to secure more attractive overall lines for a vehicle of this size. Side frames are of stress skin construction, and two longitudinal purlines to which the bulkheads are attached run the entire length of the body. These are Z-shaped sections welded to carlings and serve the purpose of increasing the longitudinal stiffness of the bulkheads.

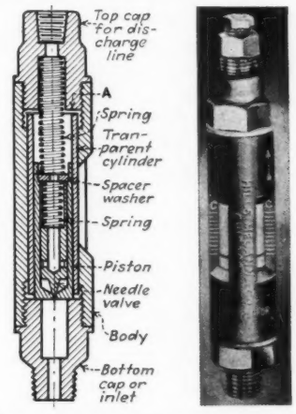
Because of sand storms, all joints are soldered, cemented or painted, and thoroughly water-tested, so there will be no possibility of dust entering the interior. The inside walls are of $\frac{1}{4}$-in. birch plywood, screwed and cemented to the battens. The wood floor is covered with linoleum and is cemented to the Phemaloid sub-floor, while the cover pan is bolted over the entire bottom surface of the body. This pan is made of 20-gage galvanized iron and is sound-deadened with cork cement. Kapok insulation is used and heaters are installed to keep the interior warm in cold weather. The stainless steel roof and sides form a reflecting surface, giving low heat-absorption and low heat losses. The trailer is completely air conditioned and individual ducts run to the upper and lower berth, the latter extending down to the edge of the partition between the seats. These ducts also serve as stiffener columns at these points.



Molded entirely of plastics, the case of the new Kadette Classic table radio is made of black Durez, capped with scarlet top and has translucent plastic grille bars. Not only is this cabinet light but it also withstands humidity and weather conditions, is peel-and-chip-proof, and will retain its original luster indefinitely

• *Registering Displacement of Metered Oil*

Because of the considerable variation in the size of drops of lubricating oils having different viscosities, it was found impossible to make an accurate determination of the amount of oil delivered to a bearing. Therefore the Hills-McCanna Company designed for their impulse type lubricator a metering sight feed which registers volumetric displacement per impulse of all oil passing through the line.



Amount of oil is measured and registered in this metering sight-feed for force-feed lubricator

As shown in the illustration, the body of the sight feed is graduated with ten lines, each line representing 1/10 c.c. of oil delivered or 1 c.c. total for the range. A red line on the piston registers through a glass tube with the graduations on the body. As the oil is delivered from the lubricator to the sight feed, the piston and valve move upward together, forcing the oil out of the upper end of the glass cylinder into the discharge line to the bearings.

For example, if each pulsation of the lubricator delivers 1/10 c.c. of oil to the sight feed, the red line on the piston will raise one graduation. As the sight feed is full of oil and the pressure above and below the piston is balanced (except for slight spring pressure at the discharge side), the red line will remain at the second graduation line. At the next impulse of the lubricator, the red line will move upward one more graduation and so on until the upper line is reached. Of course, the lubricator can be adjusted to deliver 2, 3, 4 or more c.c. for each pulsation, whereby the red line on the piston moves upward 2, 3, 4 or more graduations per impulse.

When the piston abuts the top cap at *A*, the needle valve continues its upward travel until the seat is broken because of pressure exerted on its lower face. The piston and valve mechanism then return rapidly to the bottom of the cylinder, the valve again seats and the cycle is repeated.

Trailer of 30-ton capacity, believed largest ever equipped with pneumatic tires, recently manufactured by Rogers Bros. Corporation of Albion, Pa. The overall length of the trailer is 47 ft. 9 in. having a deck width of 11 ft. 6 in. and it is entirely fabricated by arc welding

• *Synthetic Resin Transparent As Silicate Glass*

As a substitute for silicate glass, a German firm is now manufacturing "Plexiglas" made from synthetic resin, and apparently quite similar to Pontalite, a du Pont product. Whereas the best substitute for glass was thin celluloid and later, "Safety Glass" fabricated with glass and plastic laminations, the new material is used in solid form, sheets ranging in thickness from 0.020 to 0.4 in. or thicker, when desired. Having a transparency of silicate glass, this material can be used in optics for lenses; and as the weight is only one-half that of glass, it is being used in the construction of airships, airplanes, vehicles and portable apparatus. When a sufficiently thick layer is used, it is practically unbreakable as it does not possess the brittleness of ordinary glass. Withstanding shrinkage and warpage, it can be exposed to the rays of the sun and to rain without deterioration. It also resists the influences of water, steam, solutions of salts, and diluted mineral acids and alkalis. It is also resistant to oils and gasoline.

Because the heat-conductivity value of Plexiglas is less than 25 per cent that of silicate glass, its heat insulating powers are much higher and it is less liable to become misty when the outside temperature drops and the inside air is moist.

Plexiglas is also a poor conductor of electricity. The material is soft enough so that it can be cut with an ordinary wood saw. Punching, grinding, polishing and drilling are readily done and the material is of such a nature that it is workable under high temperatures in various machines used in the manufacture of synthetic resin parts.

Up to the present time, in addition to the applications already mentioned this material has been used for spectacles and spectacle frames, instrument dials, slide rules, watch crystals, goggles and glasses.

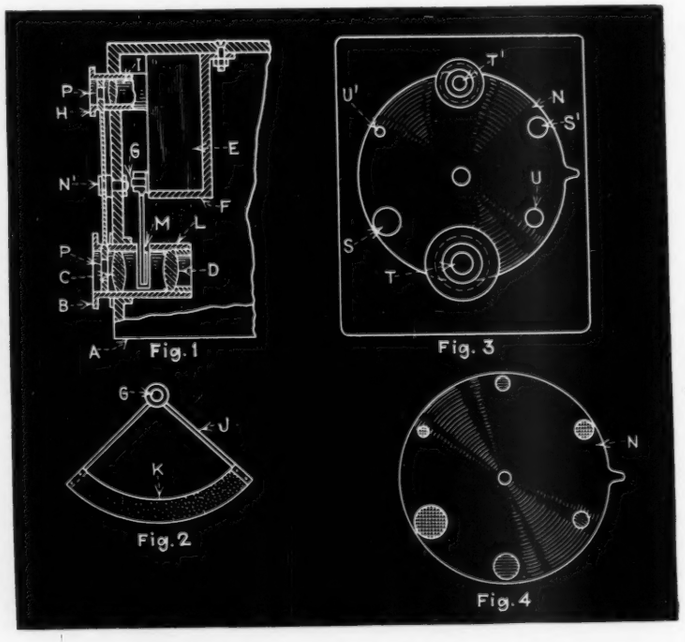
Photo-Electric Cell Used in Camera

● A camera with a device for automatically adjusting the light intensity has recently been invented and patented by Dr. Albert Einstein of Princeton, N. J., and Gustav Bucky of New York. The object of the invention is to provide means for automatically adapting the light impinging the photographic plate or cell of the camera to the light intensity of the surroundings and particularly of the object to be photographed. It also provides means for an automatic adjustment of the light intensity without necessitating the use of a power source. The camera consists of a photo-electric cell and a screen of varying penetrability to light, the screen being movable in the path of the light rays passing the camera objective.

As shown in the illustration, the camera *A* carries a photographic objective, *B* having lenses *C* and *D*. In the interior of the camera, a photo-electric cell forming a unit *E* together with a mechanism for turning a shaft is mounted and secured on the bracket *F*. The shaft *G* is adapted in such a manner as to be actuated by the photo-electric cell. The latter which receives light through the tubular fixture *H* in the front wall of the camera, with or without the help of a lens *I* in the aperture of the fixture. A frame *J* carrying the screen *K* in rigidly connected to the shaft, the screen being made of a transparent celluloid-like material. The transparency of the screen is decreased from one end to the other by shading it from clear over gray to black, as indicated in Fig. 2. This screen is so arranged that it swings in the path of the light rays passing through the objective *B* and is so adjusted that the portion of the screen having maximum transparency is in the path of the rays when the photo-electric cell is not, or only very little, energized; that is, when the light intensity is weakest. Because of the small power output of the photo-electric cell, the screen is mounted in a position between the lenses *C* and *D* the objective tube, being slitted as at *M* because the coincidence of the rays in the space between the lenses permits the screen to be made very narrow. This is important because the moving part and connection with the photo-cell unit should be made as light as possible.

For clearness, the shutter arrangement has been omitted from the drawing. When the camera is directed with its objective *B* upon the object to be photographed, the aperture or light entrance tube *H* of the photo-electric unit is directed likewise. The light intensity striking the photo-electric unit creates power which turns the shaft *G* through an angle which is a function of the light intensity. Consequently, the screen *K* is swung between the lenses *C* and *D* in such a position that the light impinging the photographic plate or film is adjusted in correspondence with the light entering the electric unit.

Means are provided for simultaneously and proportionally altering the aperture of the objective and the aperture through which the light enters the photo-electric cell. This is illustrated both in Fig. 1, 3, and 4. A disk *N* is pivotally mounted at *N'* on the front wall of the camera between the objective *B* and the tube *H*. The objective and the tube are slitted at *P* and *R* respectively, and the disk *N* passes through the slits. The disk is provided with a set of holes *S*, *T* and *U* of different sizes and with a second set of holes *S'*, *T'* and *U'*, which are so arranged that hole *T* is in front of the objective when the hole *T'* is in front of the tube. The same relation exists with regard to holes *S* and *S'* and holes *U* and *U'*. The sizes of these sets of holes are so chosen that the second one is adapted to change the aperture of the tubular fixture *H* in the same proportion as the first set is adapted to change the aperture of the objective *B*. Thus the unit *E* receives light only in correspondence with the light of the objective and properly corrects the light impinging the photographic plate. In a similar manner when using a filter for the objective, a filter of equal quality should be used for the light entrance to the photo-electric cell.



Figs. 1-4—Arrangement and details of mechanism for adjusting the exposure of a camera automatically by means of a photo cell

● *Random Jottings about New Developments*

To avoid the accumulation of greenish matter from corrosion on liquid soap dispensers, one firm is now using stainless steel for the ball valves.

* * *

Instead of flaring light-walled monel tubing for joints, a necked flange is welded to the end of the tube after slipping on a malleable iron backing flange. The face of the flange is flat and smooth and the inside corner radius is kept at a minimum. A rubber gasket is the seal.

Standards for Spring Design—I

W. M. GRIFFITH
Product Engineer, Atlas Imperial Diesel Engine Company

Working stresses, endurance limits and standard material specifications for helical spring wires

IN A previous article, "Engineering Standards for the Design of Springs" (July 1933 *P.E.*), the successive steps in the design of springs were presented. These included the selection of the material, formulas for stresses and deflections, spring properties and the equations for calculating spring proportions to avoid operation at natural frequencies, were presented. Springs made in accordance with those standards have given excellent results, although a considerable amount of difficulty was experienced owing to poor material, especially in the case of valve springs.

Since the publication of that article, tests and experiences have demonstrated that some of the maximum permissible working stresses could be modified advantageously and the Wahl correction factor incorporated in the equations for calculating the stresses, thereby obtaining a more exact design. Thus, for certain classes of springs, the working stresses should be selected on the basis of the endurance limit of the material for the stress range at which the spring is to operate, rather than using the static strength of the material in the spring equation. In establishing these endurance limits, the variation of the static strength of the material with the diameter of the wire was taken into consideration, as will be explained. Finally, to insure springs of proper strength and characteristics, and of uniform quality, materials standards were formulated and the spring materials selected and accepted according to these standards, as presented in this article. For convenience, the data and equations that appeared in the previous articles will be repeated here, some of them in modified form to bring them up-to-date.

The selection of the material and working stresses will depend largely upon the severity of service to which the spring will be subjected. To reduce this factor to a workable basis, spring service has been divided into three classes, as shown in Table I. In determining the class to which a spring belongs, care should be exercised and in case of doubt the more severe service should be assumed and the corresponding lower maximum permissible working stresses should be used in the design calculations.

In general, all springs shall be made of circular section music wire, Swedish steel, carbon steel or phosphor bronze. Chrome vanadium steel, brass, Monel metal or sections other than circular, are used only when, for some special reason, it is obviously impossible or undesirable to use one of the above materials or a circular section.

In order to control the quality of the springs in so far as possible, the Atlas Imperial Diesel Engine Company made experiments and investigations for the purpose of formulating standards covering the analyses, physical properties and acceptance tests for spring materials. Standards covering Swedish steel, carbon steel, chrome vanadium steel and music wire were formulated, covering not only the chemical composition and physical properties of the material, but also tolerances and surface conditions, and the source of supply. This necessarily required a large amount of investigation and the final results are given in the accompanying Standards Specifications for Spring Materials.

Referring to the materials standards as presented here, Class I compression springs are now made of music wire in wire sizes up to and including 0.105 in. diameter. For diameters from 0.1055 to 0.262 in. inclusive, Swedish steel is used, and for wire diameters greater than 0.262, carbon steel is used. This particular arrangement was determined upon the basis of the actual materials available in this country. Swedish steel is used for compression springs in Class I only.

Carbon steel and chrome vanadium steel are used only for the sizes above 0.2625 in. diameter, as given in the materials specifications, and also for square or rectangular wire ranging from a minimum size of $\frac{1}{32}$ x $\frac{1}{32}$ to a maximum of ½ x ½, advancing by thirty-seconds of an inch, and above ½ x ½ in., advancing by sixteenths of an inch.

Springs made of music wire should not be finished, while the other steels should ordinarily be black Ja-

Table I—Classes of Spring Service

CLASS	DEFLECTION CYCLES	STRESS RANGE PER CYCLE	SERVICE
1	Rapid, uniform stress range and continuous	Uniform from zero to maximum or intermediate to maximum	Valves, plungers, etc.
2	Rapid, varying stress range and intermittant	Variable from zero to intermediate, intermediate to maximum or zero to maximum	Governors & similar service
3	Infrequent with varying stress range or static loading	Variable from zero to intermediate, intermediate to maximum, or zero to maximum, maximum-(static)	Relief valves static loads

SWEDISH STEEL VALVE SPRING WIRE

1 — This specification covers the grade of valve spring steel to be used in the manufacture of all springs with this material specification number.

2 — **STEEL MANUFACTURE:** This steel is to be of Swedish manufacture according to approved practice by the acid open hearth or electric furnace process.

3 — **CHEMICAL COMPOSITION:**

Carbon	0.60	— 0.70	Phosphorus	0.025 max.
Manganese	0.45	— 0.65	Sulphur	0.025 max.
Silicon	0.15	— 0.25		

4 — **PHYSICAL PROPERTIES:**

Size	Tensile Properties (min.) Ultimate	Tensile Properties (min.) Elastic Limit	Torsional Properties (min.) Ultimate	Torsional Properties (min.) Elastic Limit
0.1055 and under	212,000	154,000	184,000	112,000
0.1205 to 0.1350	202,000	146,000	175,000	106,000
0.1483 to 0.1920	187,000	136,000	163,000	99,000
0.2070 to 0.2625	175,000	126,000	151,000	92,000
0.28125 to 0.34375	164,000	119,000	142,000	86,000
0.3625 to 0.4375	155,000	112,000	135,000	82,000
0.4615 to 0.5625	146,000	106,000	127,000	77,000

Reduction of area 48 per cent minimum Elongation in 10 inches 5 per cent minimum

Twist Test:

Samples taken from any part of the bundle of wire must stand 7 twists forward and reverse as shown on chart to a clean square break, the speed of twisting in air not to exceed 25 R.P.M.

Using 10 inch specimen:								
Size	0.1055	0.1205	0.1250	0.1350	0.1483	0.15625	0.1620	0.1770
Reverse Twists	23	20	20	18	17	16	15	14
Using 15 inch specimen:								
Size	0.1875	0.1920	0.2070	0.21875	0.2253	0.2437	0.2500	0.2625
Reverse Twists	20	19	18	17	16	15	15	14
Using 20 inch specimen:								
Size	0.28125	0.2830	0.3065	0.3125	0.3310	0.34375	0.3625	
Reverse Twists	18	17	16	16	15	14	14	
Using 30 inch specimen:								
Size	0.3750	0.3938	0.40625	0.4305	0.4375	0.4615	0.46875	0.4900
Reverse Twists	20	19	18	17	17	16	16	15
Size	0.500	0.53125	0.5625					
Reverse Twists	15	14	13					

5 — **SURFACE:** Upon etching with a hot solution of Hydrochloric Acid sufficiently to disclose surface defects, no seams, hairline or otherwise, pits, gouges, die marks, or other imperfections shall be revealed. Decarburization must be held to a minimum.

6 — **SIZE VARIATION LIMITS:**

0.1620 and less — plus or minus 0.0015
0.1770 and over — plus or minus 0.002

7 — **INSPECTION, REJECTIONS AND REPLACEMENTS:** All springs will be rigidly inspected at the plants as received. If more than a total of 10 per cent of the springs on any one purchase order are made of steel which fails to comply with the above specifications, or with the specifications on the detail drawing, the entire lot will be rejected. All springs rejected at the plants will be held at the seller's risk for a reasonable length of time, subject to his instructions, and shall be replaced by the seller without further cost to the purchaser.

CARBON STEEL VALVE SPRING WIRE

1 — This specification covers the grade of valve spring steel to be used in the manufacture of all springs with this material specification number.

2 — **STEEL MANUFACTURE:** This steel is to be made according to approved practice by the electric furnace or open hearth process.

3 — **CHEMICAL COMPOSITION:**

Carbon	0.60	— 0.70	Sulphur	0.025 max.
Manganese	0.45	— 0.65	Phosphorus	0.025 max.

4 — **PHYSICAL PROPERTIES:**

Size	Tensile Properties (min.) Ultimate	Tensile Properties (min.) Elastic Limit	Torsional Properties (min.) Ultimate	Torsional Properties (min.) Elastic Limit
0.1055 and under	202,000	132,000	165,000	108,000
0.1205 to 0.1350	191,000	125,000	157,000	103,000
0.1483 to 0.1920	178,000	117,000	145,000	95,000
0.2070 to 0.2625	165,000	108,000	136,000	89,000
0.28125 to 0.34375	156,000	102,000	127,000	84,000
0.3625 to 0.4375	147,000	97,000	121,000	79,000
0.4615 to 0.5625	139,000	91,000	114,000	74,000

Reduction of area 48 per cent minimum Elongation in 10 inches 5 per cent minimum

5 — **SURFACE:** Upon etching with a hot solution of Hydrochloric Acid sufficiently to disclose surface defects, no seams, hairline or otherwise, pits, gouges, die marks, or other imperfections shall be revealed. Decarburization must be held to a minimum.

6 — **SIZE VARIATION LIMITS:**

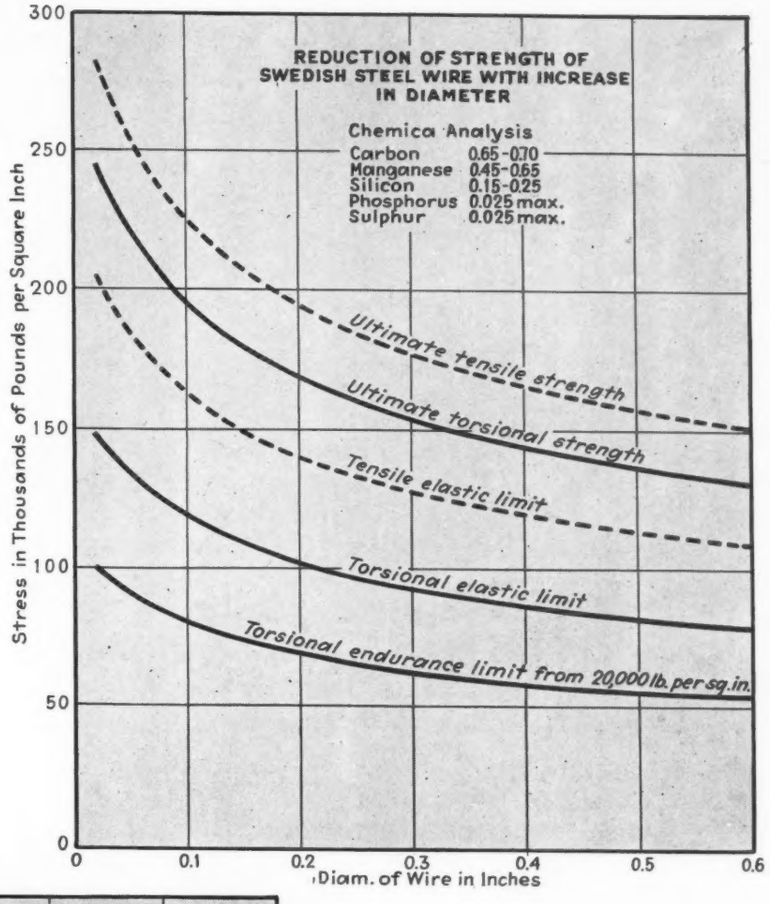
0.1620 and less — plus or minus 0.0015
0.177 and over — plus or minus 0.002

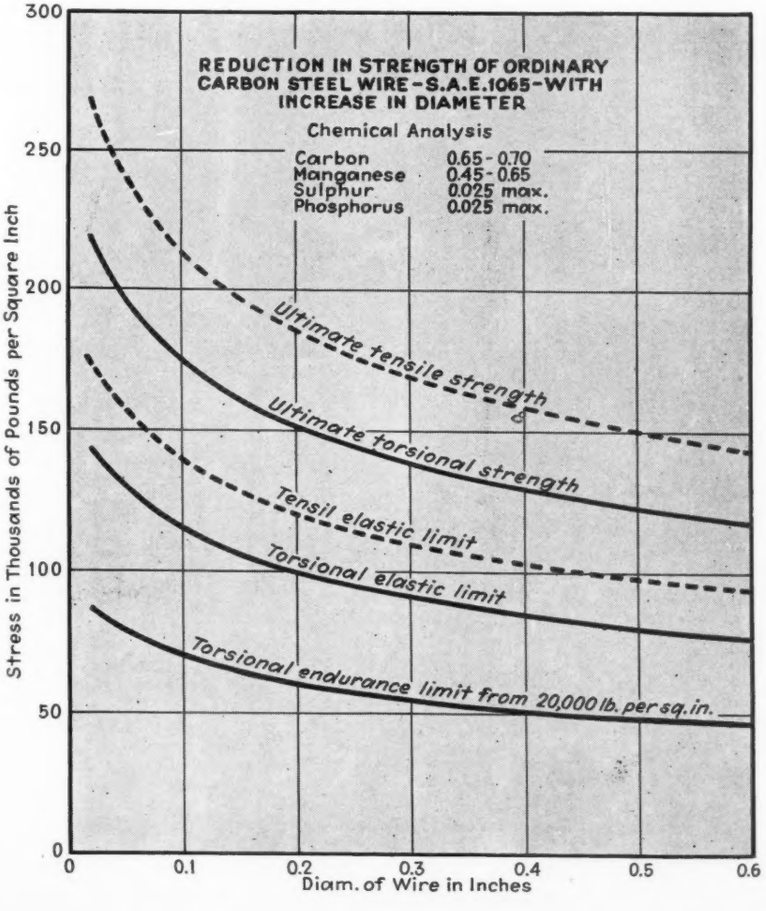
7 — **INSPECTION, REJECTIONS AND REPLACEMENTS:** All springs will be rigidly inspected at the plants as received. If more than a total of 10 per cent of the springs on any one purchase order are made of steel which fails to comply with the above specifications, or with the specifications on the detail drawing, the entire lot will be rejected. All springs rejected at the plants will be held at the seller's risk for a reasonable length of time, subject to his instructions, and shall be replaced by the seller without further cost to the purchaser.

panned. When steel springs must resist moisture, they shall be cadmium plated.

In the non-ferrous metals, SAE 81 shall be specified for phosphor bronze and SAE 80 for brass. Monel metal shall be specified as such. Phosphor bronze is used only for small springs when, depending upon the design, it appears to be the proper material for the service or when the springs must resist moisture. Of the non-ferrous materials, only phosphor bronze shall be used for springs in Class 1 service. Brass, phosphor bronze and monel metal are drawn to Brown & Sharpe gage. Square or rectangular material may be used from a minimum size of $\frac{1}{32}$ x $\frac{1}{32}$ to a maximum of $\frac{1}{2}$ x $\frac{1}{2}$ in., advancing by thirty-seconds. Springs made of non-ferrous materials should not ordinarily be finished in any manner.

In formulating the materials standards as presented here, investigations were first made to determine the tensile and torsional properties of the materials in question. All of the available literature was reviewed, in addition to which a number of prominent spring manufacturers were consulted. Thus the values indicated by the curves in Figs. 1 to 5 were established. The value of the physical properties of phosphor bronze, as indicated in Fig. 5, are based on the physical properties given in the *SAE Handbook* under Specifications SAE 81.





The physical properties indicated in Figs. 1 to 5 can reasonably be expected with the sizes of materials covered. However, in establishing the minimum physical properties designated on the materials specifications sheets, only 95 per cent of the ultimate values indicated by the charts were used. This was done to provide some factor of safety and to make certain that slight variations in the physical properties of the material would not, in any manner cause disagreement with the minimum properties indicated on the materials specifications sheets. Hence, no additional expense is involved in actually securing material with the minimum physical properties specified, as these properties are well within commercial limits.

CHROME VANADIUM STEEL VALVE SPRING WIRE—SAE 6150

1 — This specification covers the grade of valve spring steel to be used in the manufacture of all springs with this material specification number.

2 — STEEL MANUFACTURE: This steel is to be made according to approved practice by the electric furnace or open hearth process.

3 — CHEMICAL COMPOSITION:

Carbon	0.45 — 0.55
Manganese	0.50 — 0.90
Chromium	0.80 — 1.10
Vanadium	0.15 min.
Sulphur	0.5 max.
Phosphorus	0.04 max.
Silicon	0.15 — 0.30

4 — PHYSICAL PROPERTIES:

	Tensile Properties (min.)		Torsional Properties (min.)	
Size	Ultimate	Elastic Limit	Ultimate	Elastic Limit
0.1055 and under	212,000	195,000	158,000	116,000
0.1205 to 0.1350	202,000	184,000	149,000	111,000
0.1483 to 0.1920	187,000	171,000	139,000	103,000
0.2070 to 0.2625	174,000	160,000	130,000	95,000
0.28125 to 0.34375	163,000	150,000	122,000	89,000
0.3625 to 0.4375	155,000	143,000	116,000	84,500
0.4615 to 0.5625	146,000	134,000	109,000	80,000

Reduction of area 48 per cent minimum
Elongation in 8 inches 3½ per cent minimum
Rockwell " C " 42–46

5 — SURFACE: Upon etching with a hot solution of Hydrochloric Acid sufficiently to disclose surface defects, no seams, hairline or otherwise, pits, gouges, die marks, or other imperfections shall be revealed. Decarburization must be held to a minimum.

6 — SIZE VARIATION LIMITS:

0.1620 and less — plus or minus 0.0015
0.177 and over — plus or minus 0.002

7 — INSPECTION, REJECTIONS AND REPLACEMENTS: All springs will be rigidly inspected at the plants as received. If more than a total of 10 per cent of the springs on any one purchase order are made of steel which fails to comply with the above specifications, or with the specifications on the detail drawing, the entire lot will be rejected. All springs rejected at the plants will be held at the seller's risk for a reasonable length of time, subject to his instructions, and shall be replaced by the seller without further cost to the purchaser.

MUSIC WIRE SPRING STEEL

1 — This specification covers the grade of valve spring steel to be used in the manufacture of all springs with this material specification number.

2 — STEEL MANUFACTURE: This steel is to be of Swedish manufacture according to approved practice by the acid open hearth or electric furnace process.

3 — CHEMICAL COMPOSITION:

Carbon	0.60 — 1.00
Manganese	0.25 — 0.50
Silcon	0.10 — 0.20
Sulphur	0.025 max.
Phosphorus	0.025 max.

4 — PHYSICAL PROPERTIES:

	Tensile Properties (min.)		Torsional Properties (min.)	
Size	Ultimate	Elastic Limit	Ultimate	Elastic Limit
0.008 and under	363,000	216,000	297,000	163,000
0.009 to 0.012	360,000	214,000	295,000	162,000
0.013 to 0.020	346,000	207,000	285,000	156,000
0.022 to 0.030	334,000	201,000	275,000	150,000
0.032 to 0.040	324,000	195,000	266,000	145,000
0.042 to 0.051	313,000	188,000	256,000	141,000
0.055 to 0.063	303,000	181,000	248,000	136,000
0.067 to 0.078	292,000	175,000	238,000	130,000
0.082 to 0.090	283,000	170,000	232,000	126,000
0.095 to 0.105	275,000	164,000	225,000	123,000

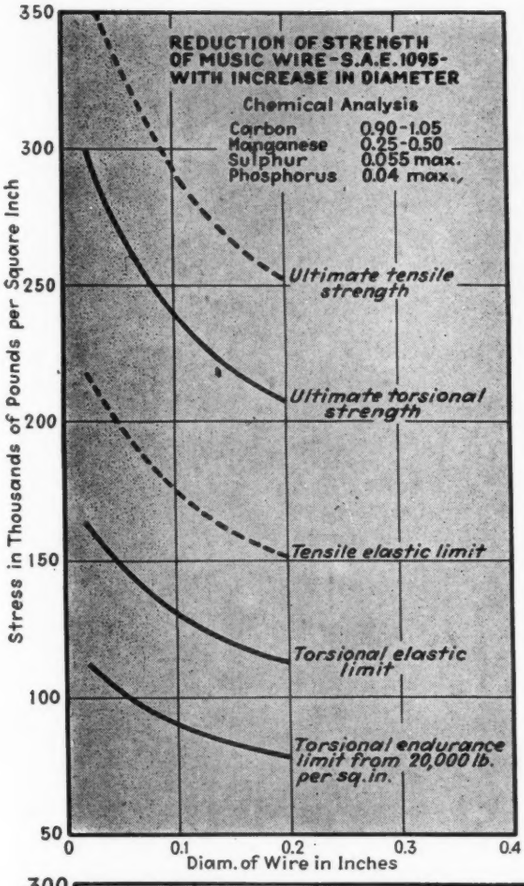
Reduction in area 46 per cent minimum
Elongation in 8 inches 2 per cent minimum

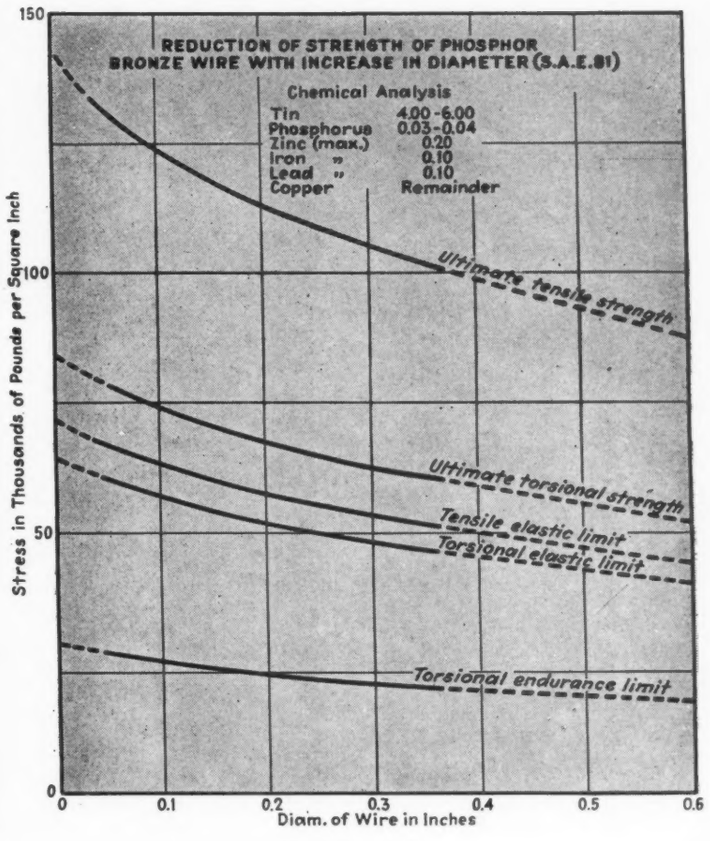
5 — SURFACE: Upon etching with a hot solution of Hydrochloric Acid sufficiently to disclose surface defects, no seams, hairline or otherwise, pits, gouges, die marks, or other imperfections shall be revealed. Decarburization must be held to a minimum.

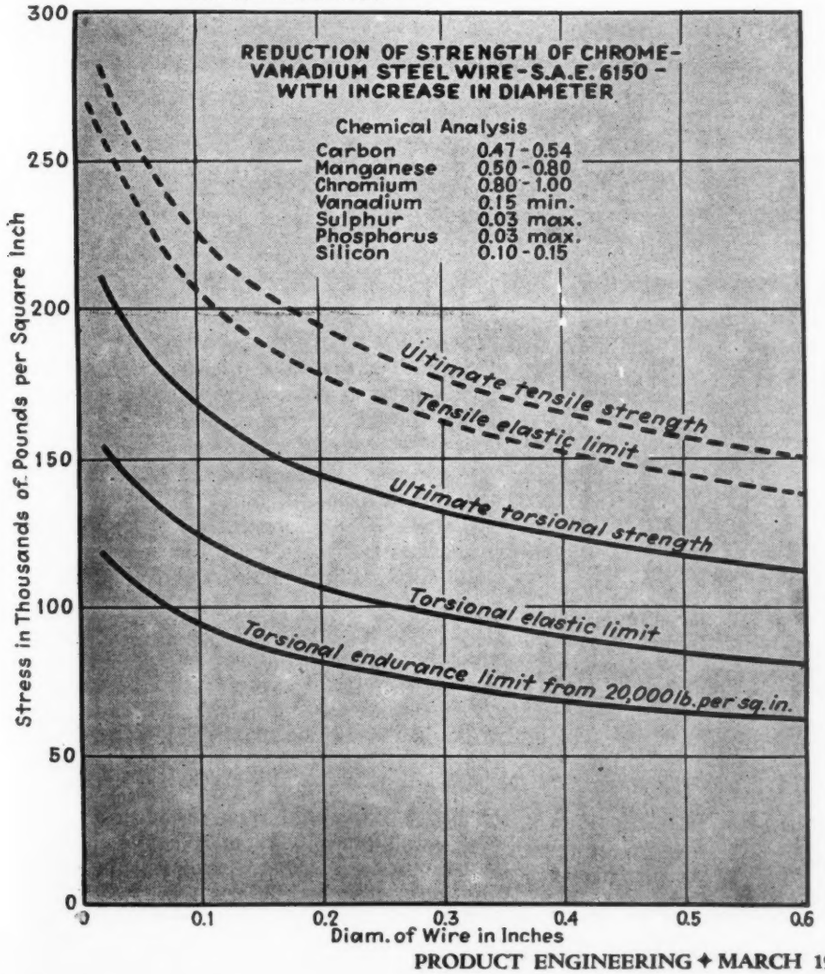
6 — SIZE VARIATION LIMITS:

0.025 and under — plus or minus 0.00025
0.027 to 0.063 — plus or minus 0.0005
0.067 and over — plus or minus 0.001

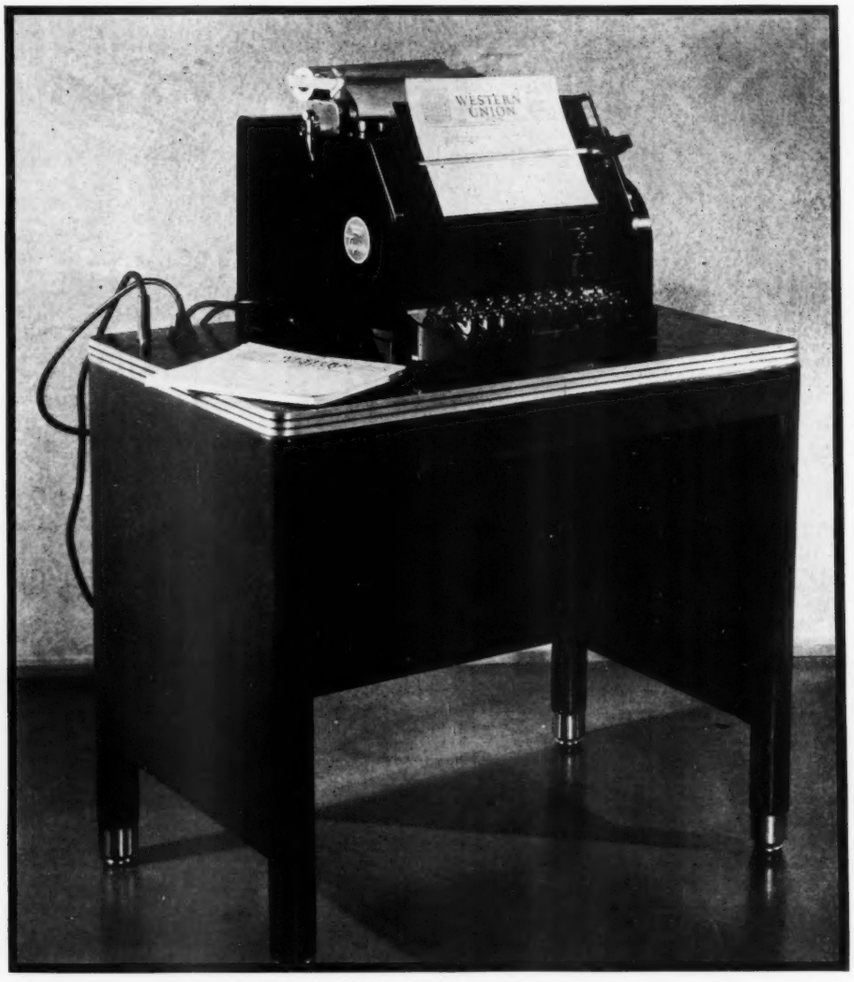
7 — INSPECTION, REJECTIONS AND REPLACEMENTS: All springs will be rigidly inspected at the plants as received. If more than a total of 10 per cent of the springs on any one purchase order are made of steel which fails to comply with the above specifications, or with the specifications on the detail drawing, the entire lot will be rejected. All springs rejected at the plants will be held at the seller's risk for a reasonable length of time, subject to his instructions, and shall be replaced by the seller without further cost to the purchaser.


REDUCTION OF STRENGTH OF MUSIC WIRE - S.A.E. 1095 - WITH INCREASE IN DIAMETER
Chemical Analysis
Carbon 0.90-1.05
Manganese 0.25-0.50
Sulphur 0.055 max.
Phosphorus 0.04 max.
Ultimate tensile strength
Ultimate torsional strength
Tensile elastic limit
Torsional elastic limit
Torsional endurance limit from 20,000 lb. per sq. in.
Stress in Thousands of Pounds per Square Inch
Diam. of Wire in Inches

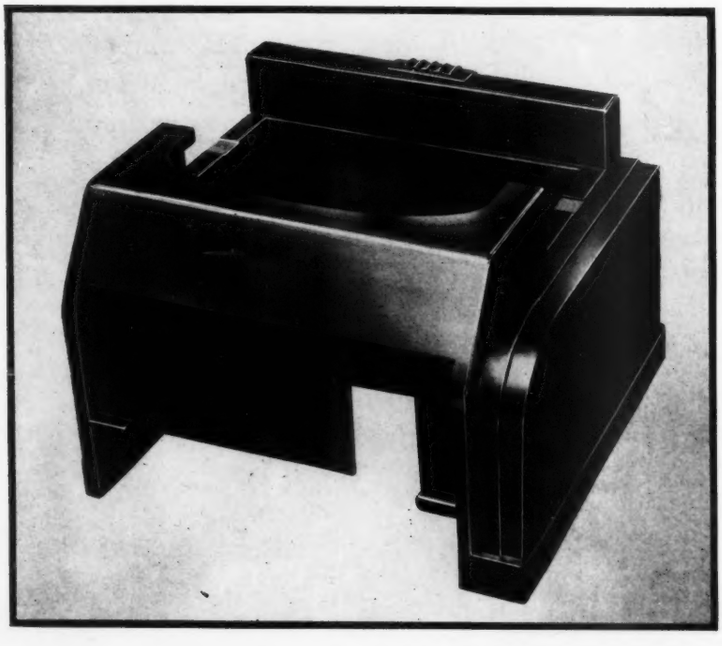



REDUCTION OF STRENGTH OF CHROME-VANADIUM STEEL WIRE - S.A.E. 6150 - WITH INCREASE IN DIAMETER
Chemical Analysis
Carbon 0.47-0.54
Manganese 0.50-0.80
Chromium 0.80-1.00
Vanadium 0.15 min.
Sulphur 0.03 max.
Phosphorus 0.03 max.
Silicon 0.10-0.15
Ultimate tensile strength
Tensile elastic limit
Ultimate torsional strength
Torsional elastic limit
Torsional endurance limit from 20,000 lb. per sq. in.
Stress in Thousands of Pounds per Square Inch
Diam. of Wire in Inches

The second part of this article, which will appear in the next number of *Product Engineering*, will cover endurance limits charts for the various spring materials, the selection of permissible working stresses, and formulas for the design of helical springs, incorporating the Wahl factor.



MODERN MECHANICAL DESIGNS—

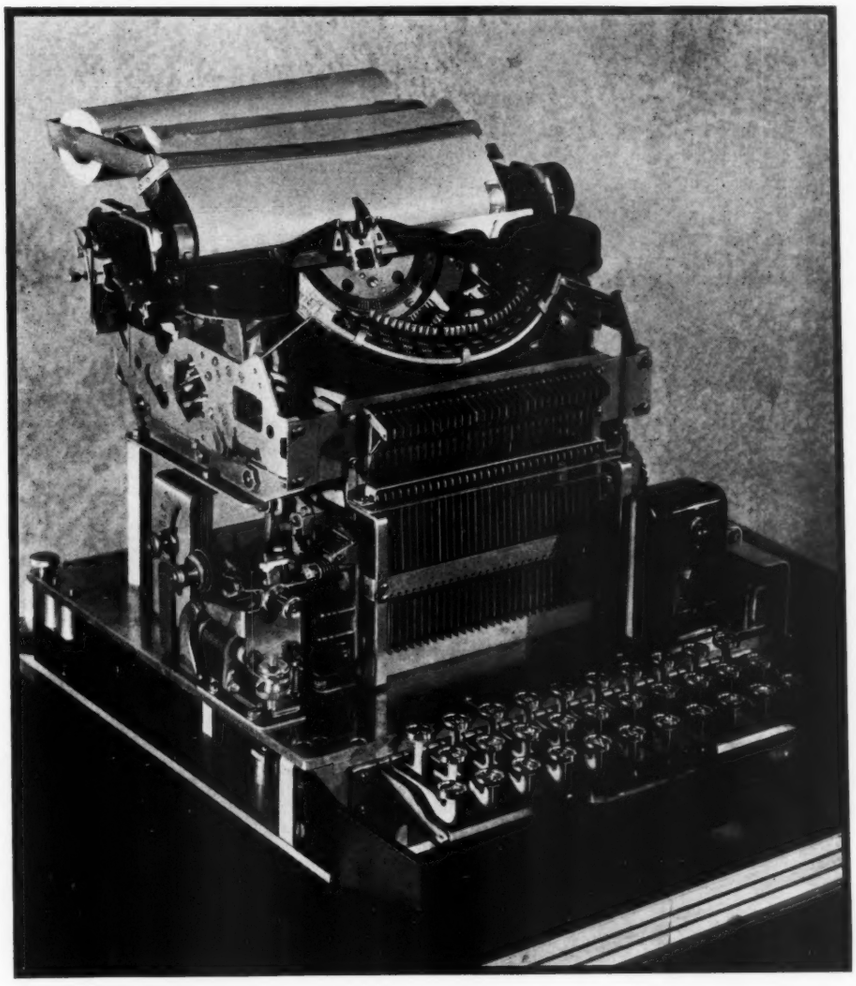


HERE is a product that is not sold but is furnished to customers who are buying service. Yet the value of attractive appearance was great enough to be the dominating design motive for the cover of the Western Union Telegraph Company's high-speed telegraph printer. Noise isolation was an equally important consideration.

Because of its inherent good appearance, permanent lustrous finish, light-weight, adequate strength and noise absorbing property, molded Textolite with wood filler was used for the cover. In addition, a surface design could be had in a molded piece that would be too costly to produce in metal.

As finally designed, the finished hood weighs 6 lb., 13½ oz., is 14½ in. wide, 15 in. long and 10¼ in. high. The housing is simply for protection and decoration, it is not an integral or structural part of the printer.

To permit changing the ribbon and paper the cover is open on top. To prevent the sound of the electrically driven machine, which

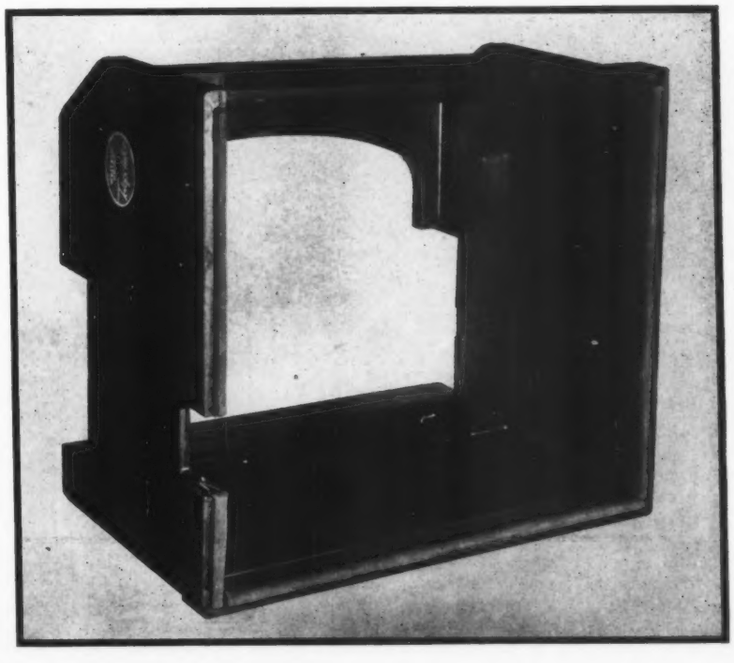


THE W.U. TELEGRAPH PRINTER

was more than was desirable, from getting out into the room, further noise isolation was obtained by padding the inside of the cover with felt and rubber, and by covering the bottom edge which rests on the desk with a rubber strip.

From a tentative design, the Western Union engineers made a wooden scale model which was turned over for suggestions to the engineers of the General Electric Company, which company molded the covers. A few minor changes were made primarily to eliminate mold parting lines where they would detract from the good appearance of the piece.

In making the molds, which were designed by General Electric Company engineers, shrinkages were used so that other plastic compounds could be molded if desired. Six months elapsed from the time that the initial design was completed until a sample cover was submitted. The total time to get it into production was one year.



Rubber Mountings

for Large Industrial Machines—II

STUART H. HAHN

Physical Research Laboratory, The B. F. Goodrich Company

Vibrograph records of two heavy machines show the effectiveness of rubber mountings in isolating vibrations. Factors of design and characteristics of rubber shear mountings for machinery are also considered

IN EVERY CASE where a direct comparison has been possible, the effectiveness of rubber mountings in reducing the transmission of vibration from machinery installed in the plants of the B. F. Goodrich Company has been striking. Several of these installations were illustrated and described in Part I of this article (February *P.E.*). No delicate instruments are required to make such comparisons, one's sense of hearing and touch reveal the marked contrast between mounted and unmounted machinery of the same type. However, as a means of recording and measuring the effectiveness of the mountings a number of vibrograph records have been made from time to time, Figs. 8 and 9 show typical records.

The floor and foundation vibration records were made either by a single-element, inertia-type instrument having a magnification ratio of about 27:1 or by a Shrader Tri-dimensional vibrograph having a magnification ratio of about 125:1. The single-element instrument was designed and built in this laboratory. The Shrader instrument employs a photographic system for recording three mutually perpendicular components of the relative motion between the frame of the instrument and an inertia mass supported by helical coil springs.

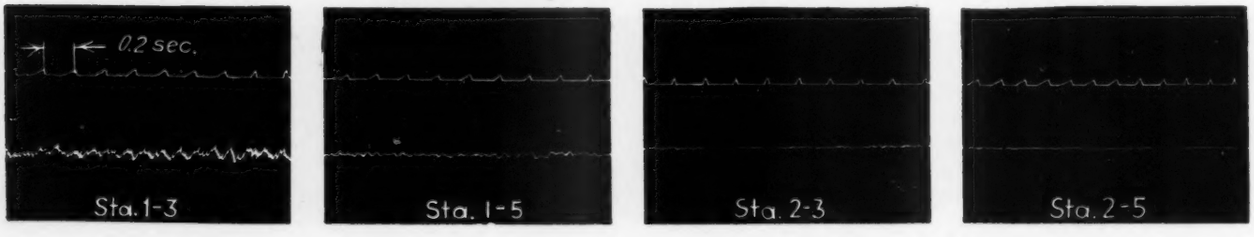
Fig. 8 shows a comparison between the floor vibrations set up by two similar Gordon Plasticators as measured with a single-element vibrograph at four stations adjacent to each of the two machines shown in Fig. 9. Machine No. 2 is that supported by flat plate shear mountings, the other machine being bolted and grouted to the floor. The two machines are in adjacent bays of the sixth floor of an eight-story, heavily constructed reinforced concrete building having 12-in. floors. During the test of each machine the other was shut down, other machinery in remote parts of the building producing no more than a barely measurable amplitude of vibration.

The reduction in amplitude of vibration shown by these records is of the order of 75 to 90 per cent. Note particularly the reduction of higher frequency vibrations. The vibration set up by the unmounted machine, in one region near the wall of the building, is severe enough to cause a very distinct relative movement between the floor slab and the wall although the wall be supported between building columns by the slab.

Fig. 10 shows a similar set of records taken near the 20-in. Gordon Plasticator. Records were made at each of the several stations with the Tri-dimensional vibrograph while the machine was being operated in the normal manner on its mountings and again while it was connected rigidly to the floor.

The earliest use of rubber as a spring device was undoubtedly some unrecorded application of bands or cords of native unvulcanized rubber in tension. Prob-

Fig. 8—Vibrograph records showing typical vibration near fifteen-inch plasticators; stations 1-3 and 1-5 are for No. 1 machine which has no rubber mountings, and stations 2-3 and 2-5 are for No. 2 machine supported by sandwich-type rubber shear mountings



ably the earliest well-known record of the use of rubber as a mechanical cushioning device was its use in compression for railway buffers (Reference 5). Since the beginning of the rubber industry rubber pads, blocks, and special molded forms stressed in compression have supported vibrating loads or have cushioned machines and vehicular bodies from external shock and noise. It has been only within comparatively recent years that the scientific approach to vibration problems has eliminated the older cut-and-try methods of designing rubber parts used for such purposes.

Many technical reasons have influenced the present trend toward the almost exclusive use of shear-type

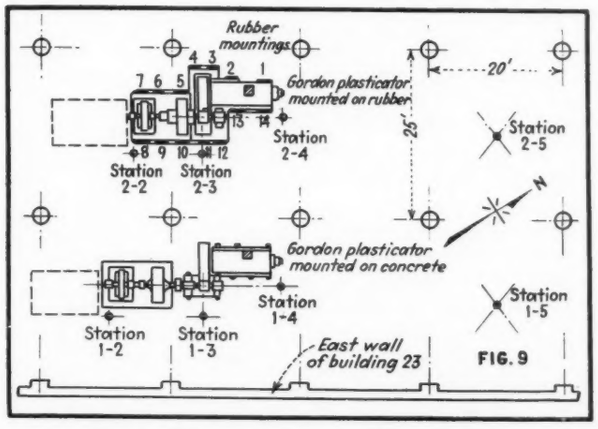


Fig. 9—Plan showing location of vibrograph stations and position of plasticators

rubber mountings in preference to all other types. The principal reason is that rubber is much more readily deformed in shear than it is in compression or tension as may be seen from Fig. 11 (Reference 6). The most compact form of the shear-type mounting is the tubular type, especially that used in mounting the machines previously mentioned and shown in Fig. 12. This type has many practical advantages, the chief of which are:

1. The ability to carry relatively high shear stresses at the rubber-metal interface of its smaller inner metal member.
2. Adaptability to a housing structure which tends to reduce the danger of oil contamination of the rubber.
3. The simplicity and low cost of the metal parts.
4. The ease with which an adjustment can be made.

Special or combination types of shear mountings have been made to fit particular products. In many of these the weight of the supported machine is carried by compressive or tensile stresses in the rubber while its elasticity in shear is used to secure the desired cushioning of vibration.

A brief review of certain of the characteristics of rubber as an engineering material may clarify some of the problems which arise in conjunction with its use as an elastic support for industrial machinery. Soft rubber is differentiated from the more usual materials of construction chiefly by the comparatively large deformations to which it may be subjected within its elastic range.

Rubber has no clearly defined elastic limit. Its breaking strength in tension based on the initial cross sectional area ranges from about 1,000 to about 5,000 lb. per sq.in. High-grade compositions suitable for mounting service show a breaking strength of about 3,500 lb. per sq.in. at an elongation of from 500 to 900 per cent, depending on the hardness of the compound. The tensile load-deflection relation and the shape of the load-deflection curve depend to a great extent on the composition of the rubber. For relatively high gum compounds it is roughly S-shaped.

Upon being allowed to recover from its deformed shape, a piece of rubber will return nearly but not quite all of the energy delivered to it during the initial deformation. This energy loss in each cycle of deformation, which results from hysteresis or internal friction, varies over a wide range, depending again on the composition of the rubber. For the types of rubber developed for mounting service this loss is relatively small and the temperature rise resulting from the normal vibratory movements of the mounting will not be perceptible.

The small hysteresis effect inherent in rubber mountings is of value in serving to damp out extreme vibratory movements in the range of resonance through which every elastically mounted machine must pass when starting or stopping, although it does increase the transmission of vibration in the normal range of operating speeds by a small amount. Rubber is superior to other organic materials in having small

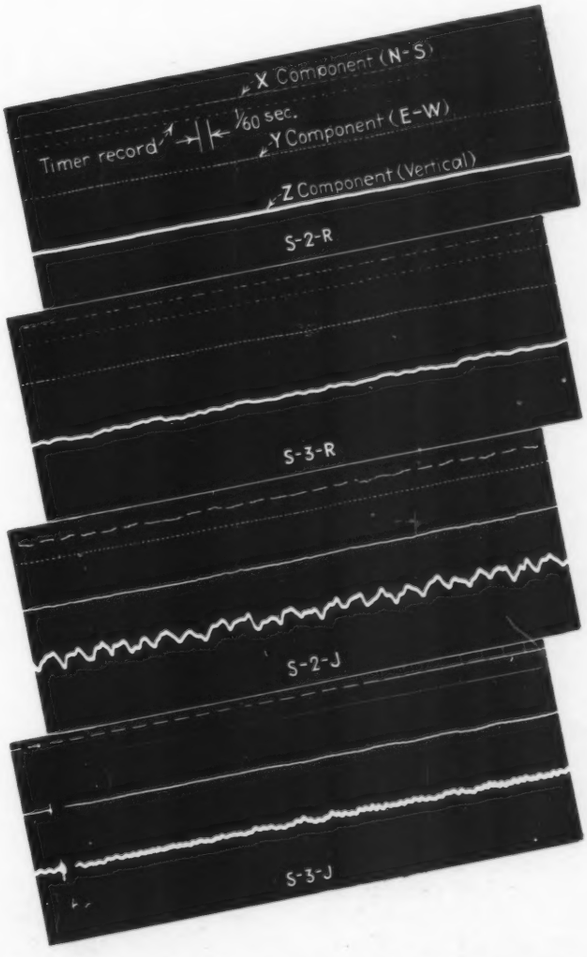


Fig. 10—Floor vibrations near twenty-inch plasticator with and without rubber mountings; vibrograph station positions similar to No. 2 and 3 shown in Fig. 9, vibration shown in S3R record exceeds that shown in S2R because of change in operation of heavy machinery on ground floor

hysteresis and it is inferior in this respect only to steel among the common elastic supporting media (Reference 7). It is well known, however, that auxiliary damping means are usually required to permit safe operation of steel spring mountings.

Coupled with hysteresis are the related effects of creep and permanent set. The deformation of rubber under constant load will change slowly with time at a rate depending on temperature, type of compound and its state of vulcanization, and on the degree and duration of stress.

A graph, Fig. 13, shows the average deflection versus time relation for the tubular shear-type mountings which support a pair of 60-in. rubber warming mills and their drives. The time scale is shown on a logarithmic basis because the creep rate decreases with extreme rapidity during the early portion of the time after application of load to the mountings. The nearly linear relation between deflection and the logarithm of time holds for about 100 days after application of load. After the first 100 days, the relation has been, up to the time this paper was prepared, nearly linear with time in every case which the author has studied. This condition results in an upward turn

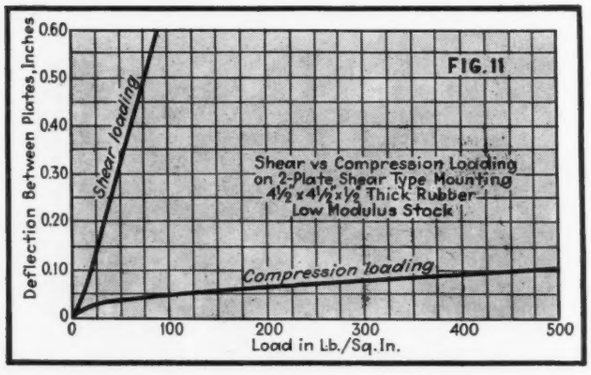


Fig. 11—Shear versus compression loading of rubber

of the latter portion of the logarithmic graph. If this portion were plotted on a linear time scale, it would be nearly a straight line showing an increase in deflection of about 7½ per cent of the initial deflection per year.

A part of the deformation of rubber which results from creep is recoverable and a part of it lost. The recovery of form after complete removal of load follows somewhat the same time relation as does creep under load. The "permanent set" of a rubber body is a measure of its permanent change from its initial form and is usually measured at a specified time after removal of a given load from a standard test piece which has been held at a controlled temperature. Permanent set varies over a large range for different rubber compounds.

The irregularities shown in the above creep curve are caused chiefly by a temperature effect which is more or less peculiar to rubber. The deflection of rubber under constant load usually decreases with increase in temperature. Thus a rubber band supporting a moderate load will shorten as its temperature is raised. The degree of the effect depends principally on the stress in the rubber and type of compound employed.

The properties of rubber are varied by proper selection and incorporation of reinforcing, loading, or softening pigments, and by vulcanizing agents together with the choice and treatment of the crude rubber. The properties depend also on the proper control of vulcanizing conditions and their degree of permanance is related to the conditions of service and use of small quantities of certain auxiliary materials such as the age resistors. The interrelation between the various properties is such that specifications of the characteristics of a compound will usually be limited by the choice of a desired principal property. For instance, it is usually difficult to secure very low hysteresis in relatively hard or stiff rubber compounds.

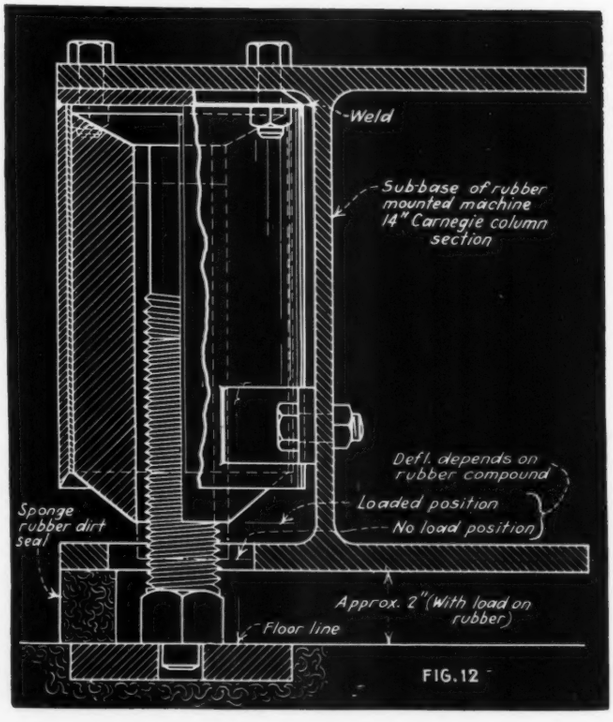
A series of standard shear mountings, as shown in Fig. 14, has been made available to the trade for general purpose uses. These cover a fairly wide range of load and deflection capacities.

The Design of Rubber-Mounted Assemblies

The application of elastic mountings of any type to a machine will be facilitated by careful study of a few fundamental design factors. In other words the complete assembly must be more than a simple combination of a suitable number of rubber mountings, properly designed and located, and a normal machine designed to be mounted rigidly.

In most installations the primary requirement is that of having the base or frame stiff enough to maintain proper alignment of all parts relative to each other. Any contribution to such rigidity offered by the foundation or floor on which the rubber mountings rest is not considered. The most economical design for rubber mountings will usually be one which has been worked out expressly for that purpose.

Fig. 12—Cross-section of tubular shear unit



However, as with the rubber machinery units which have been described, it is often possible to work out an economical welded steel sub-base. The mill and Plasticator units probably required heavier sub-bases than those necessary for most types of machinery. Many other machines will need no special sub-base or reinforcement because of the relatively light dynamic loads imposed or because it may be relatively unimportant to maintain exact alignment between the several parts fo the machine.

It is always desirable, in the interests of economical design and machine stability, to avoid the presence of extremely low disturbing frequencies in the rubber-mounted machine, that is, frequencies lower than about 10 or 12 cycles per second. It should be recalled that the lowest disturbing frequency which is to be isolated should be at least four times the fundamental or resonant frequency of the elastically supported machine (Reference 3).

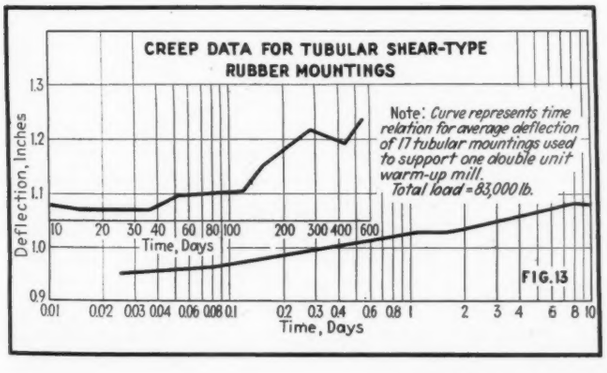
In the usual case in which the stiffness of the springs has a constant value the resonant frequency is inversely proportional to the square root of the static deflection of the elastic support, that is expressing static deflection as d in in. and resonant frequency as f_r in cycles per sec.

$$f_r = \frac{3.13}{\sqrt{d}} \quad \text{(Reference 3).}$$

The error in f_r resulting from the assumption of a linear modulus for shear mountings in this equation will be less than 10 per cent; that is, f_r (calculated) will be more than the actual value by not more than 10 per cent. For very effective isolation of a disturbing frequency the ratio of impressed to resonant frequencies should be 8 or more, although for rubber mountings the transmissibility for a ratio of 4 is only about 0.075 and further increase in the ratio results in only a very gradual decrease in transmissibility. In other words, a rubber mounting will transmit only about 7½ per cent of an impressed vibratory force having a frequency four times the resonant frequency.

One of the primary advantages of rubber as an elastic support is its relatively low transmissibility of sound, a benefit which can be lost by the careless direct connection of a fairly rigid pipe line or conduit to a rubber-mounted machine. In fact, the noise nuisance created by such a connection is likely to exceed that produced by a similar solid connection to a rigidly mounted machine. A unitary arrangement of the power source on the same base with the driven machine is often possible.

Fig. 13—Creep data for tubular shear-type rubber mountings


CREEP DATA FOR TUBULAR SHEAR-TYPE RUBBER MOUNTINGS
Note: Curve represents time relation for average deflection of 17 tubular mountings used to support one double unit warm-up mill. Total load=83,000 lb.
Deflection, Inches
1.3
1.2
1.1
1.0
0.9
10 20 30 40 60 80 100 200 300 400 600
Time, Days
0.01 0.02 0.03 0.04 0.06 0.08 0.1 0.2 0.3 0.4 0.6 0.8 1 2 3 4 6 8 10
Time, Days
FIG. 13

Variable or shifting loads should be avoided if possible, especially on machines carried by very soft mountings. A shift in the center of gravity of the machine will cause it to tilt and produce non-uniform loading of the various mountings in the group. An increasingly more severe off-level condition will then result because of creep variations among the various mountings.

Often it is best, in the interests of stability and reduction of excessive rocking movements of a rubber-mounted machine, to avoid placing the mountings too low in relation to the center of gravity. The ideal arrangement calls for the location of the elastic support

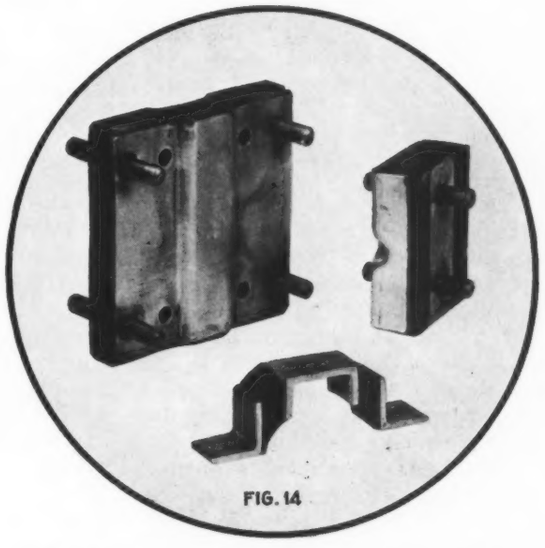


Fig. 14—Types of standard rubber mountings

at or near the center of gravity of the complete assembly. Since such an arrangement is seldom practical, the most satisfactory solution with multiple mountings will be to raise the horizontal plane of the mountings as far as seems advisable. An improvement in dynamic conditions will be obtained in a few cases by lowering the horizontal plane of the mountings so as to give the machine a higher polar moment of inertia, thus removing one or more of its resonant frequencies of rocking from those critical regions determined by the frequencies of the vibratory forces to be isolated.

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STYLE Plus Performance

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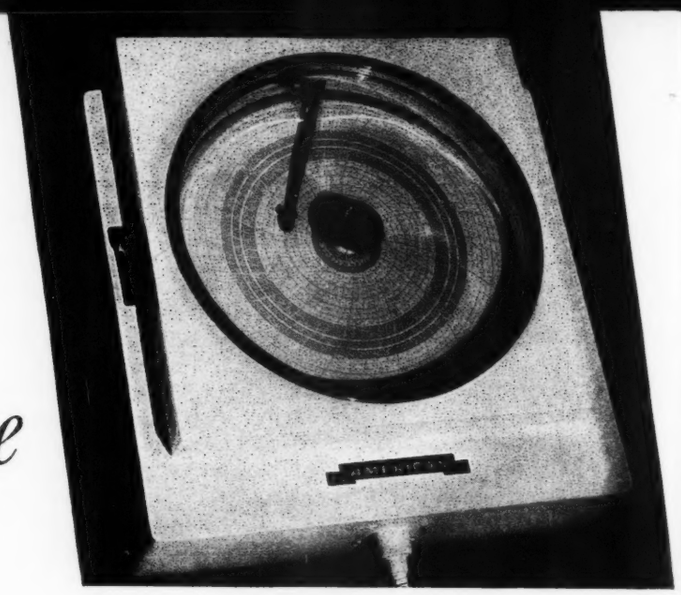


Thermometer as redesigned to harmonize with dairy equipment

INDUSTRIAL designing creates style by blending color, symmetry, and simplicity of form with fundamental engineering and functional design. It aims to give an industrial product a pleasing appearance consistent with duty.

Good style and efficient mechanical performance in a product add intrinsic value by creating the desire for ownership, by intensifying the pride of possession, by increasing the satisfaction and pleasure derived from operation, and by establishing a higher standard of accomplishment. Fashioned through the mediums of color, form and finish, and through the utilization of new processes, methods and materials, everyday commercial products are transformed into sleek, shining, graceful articles.

Industrial designing should satisfy functional and utilitarian requirements, as well as add style; it should do more than apply a mere sugar-coating of superficial design gloss to hide mediocrity or inherent functional, mechanical, or material defects. It should not be characterized by the extreme or bizarre in appearance. For example, streamlining, which is functionally desirable when applied to airplanes or automobiles, is reduced to an absurdity when applied to household appliances or furniture. The excellence of an industrial design is judged by the degree to which simplicity, appearance, and space economy are combined with mechanical efficiency and functional design consistent with purpose and cost.

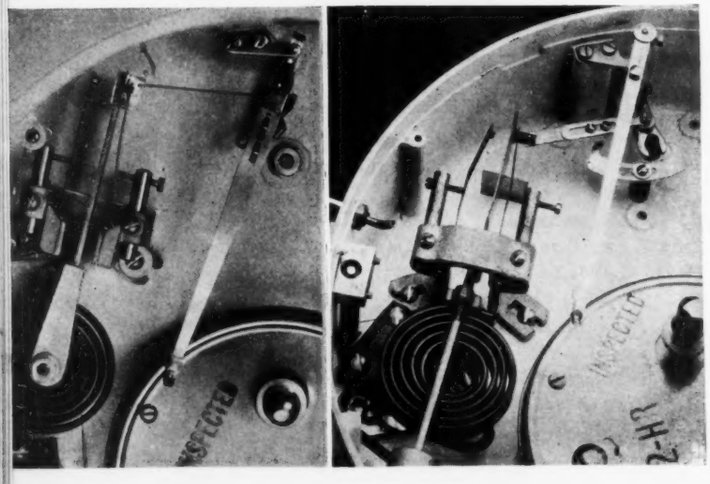
The ideal combination of style and utility will be approached when the industrial and design engineers together work out each step and detail in the design of a new product. Choice of basic form and selection of materials for any product should be dictated by sound engineering and full consideration of the problems and requirements of production and sales. But the industrial designer should be permitted as much freedom as possible in utilizing the latest developments in metals, plastics, alloys, and finishes, and should not be required to style a product which is already completely designed.

Along with the natural increase in sales value are other beneficial results and distinct advantages which show up directly in savings in manufacturing and sales that often will offset the additional cost of industrial designing. Factory workers take a personal pride in a well styled product, and workmanship and quality will be better than in a comparable product which is not styled. This relationship between quality and appearance holds throughout the entire organization, and is manifested as pride of company achievement. In setting a higher appearance standard industrial designing also tends to raise the standard of functional design and performance.

This coordination of industrial and design engineering is well illustrated by reference to an actual case history covering the redesign of a highly special dairy thermometer used to record the temperature and time of milk Pasteurization. The present day trend in the design of dairy equipment is toward simplicity of form and the use of modern materials and finishes to conform with requirements for cleanliness and sanitation. Recognizing this, it was decided to replace a current design with a new model of pleasing appearance, which would harmonize with other modern dairy equipment, and which would establish new standards of performance and serviceability.

Nine distinct steps were taken to develop the prod-

Illustration to left shows new mechanism for operating pen arm which indicates Bourdon tube deflection. Old mechanism is at right

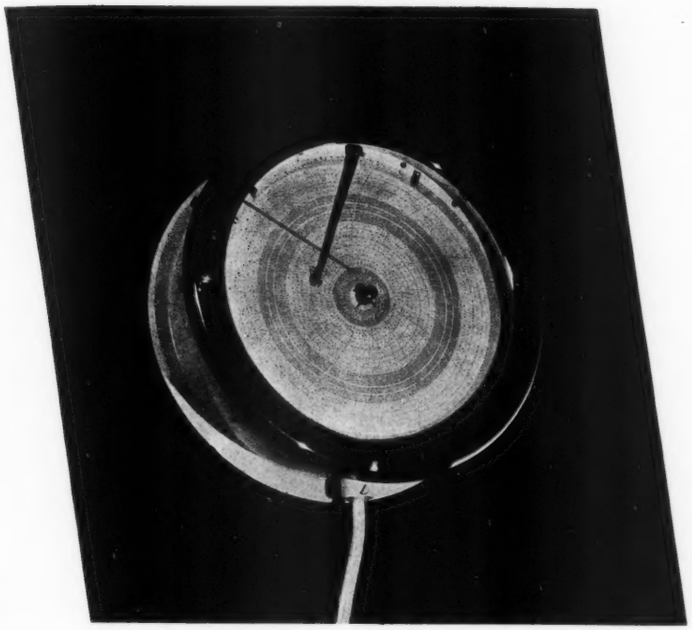


uct and place it on the market. A specification sheet covering performance and materials was prepared by the company's design and metallurgical engineers.

The industrial designer and the company engineers coordinated in the preparation of various designs, from which several were tentatively selected by a "Design Board" representing management, engineering, production and sales.

Models were made of the selected designs as a visual check of the industrial design. From these models a final composite design was formulated and approved by the "Design Board."

An experimental working model was then made which incorporated all features of industrial and design engineering. This model was then subjected to



Old instrument in conventional round case which was superseded by rectangular case of cast aluminum

a break-down test to determine its probable life under actual service conditions; changes revealed as necessary by this test were incorporated in a final model which in turn was approved by the "Design Board."

The next step consisted of the engineering, design and drafting incident to the preparation of complete working drawings and specifications.

A specified number of complete instruments were built by the production department before work was started on production tools. These instruments were placed in actual service, and carefully checked over a period of time for performance and reliability. By making the instruments before final tooling, the production department became familiar with the product, had a chance to check the accuracy of the drawings, and had an opportunity to try out the design from a manufacturing point of view. By placing the custom-built instruments in actual service, the usual "bugs" were detected and eliminated, thus avoiding expensive tool changes and additions after the instrument was in production.

After the design proved to be satisfactory a program was undertaken covering tooling, working out of production problems, engineering supervision, training of workmen, and the handling of the usual routine details incident to the starting of production of a new product.

The new product was then presented to the company's own sales force and to the trade; this presentation consisted of trade journal advertising, publicity, sales meetings, and direct sales work with the customer.

The last step was a continuous check by the sales and engineering departments on customer's acceptance. The field and service history of the new product was carefully followed. This control assisted the detection of minor defects which were then corrected before serious functional or sales handicaps developed.

The story of the industrial design is told graphically by the illustrations which show the exterior and interior views of the old instrument in the conventional round case, in contrast to the same views of the new and improved instrument in the modern rectangular case. The case and cover are of cast aluminum, finished with a highly polished moisture resistant air-dried white enamel, and tightly sealed to keep the interior dry, even in the excessively humid atmosphere of a dairy.

The bulb of a dairy recorder must be removed daily, cleaned and sterilized with live steam, and replaced, ready for the next day's run. The capillary tubing must be flexible enough to permit easy handling and must withstand constant flexing and bending year in and year out without failure. All parts of the bulb and capillary which come in contact with the milk must be corrosion-resistant and of an alloy which will not contaminate the milk.

The required combination of strength, flexibility, and corrosion resistance is obtained by using a stainless steel bulb and a stainless steel interconnecting capillary. To provide strength the capillary is covered with a stainless steel spiral armor held together with a stainless steel wire interlock. This armor, in turn, is covered with a stainless steel braided armor to form a smooth surface. For additional protection against breakage, an additional short length of stainless spiral armor is provided at the case and at the bulb—the two points where breakage would be most likely to occur as a result of the frequent handling of the bulb which must be cleaned after each batch.

As further insurance against corrosion, all welded connections between the capillary, bulb, and Bourdon spring are made with the atomic hydrogen welding process.

The entire pen arm movement was redesigned to reduce weight, inertia, and friction. Likewise, the Bourdon spring mounting, and the retard spring bracket were redesigned for simplification and better functional performance. The pen arm movement is entirely of stainless steel, and the pen itself is of glass. The connecting link is chromium plated phosphor bronze, and the retard springs are beryllium copper to obtain the highest degree of corrosion resistance combined with high fatigue life. All other miscellaneous parts are heavily plated, and the clock is sealed against moisture.

It is interesting to note that the introduction of the new instrument in a highly competitive market was followed by an immediate and continued increase in sales volume; and also that service and repair costs on the new instrument were almost entirely eliminated.

Principles of Design for NON-CIRCULAR GEARS

PART IV—Kinematics of Elliptical Gears

Angular displacements, velocities and accelerations in single pair and linear trains of elliptical gears with a method for balancing inertia torque by means of spring elements

J. J. PESQUEIRA

TO DEVELOP a method which can be used to determine the angular displacement of elliptical gears, it is necessary to establish an expression that evaluates the lengths of the contact radii at any instant during the cycle.

Assume a pair of equal rolling ellipses L and L', engaging each other, as shown in Fig. 13, the axes of rotation being through the foci. The respective contact radii of these curves at any instant are

$$OQ = r = \frac{p}{1 - e\cos\theta} \text{ for } L \qquad (47)$$

$$O'Q = r' = \frac{p}{1 + e\cos\theta'} \text{ for } L' \qquad (48)$$

where p is the semi-latus rectum and e the eccentricity of the ellipses, θ and θ' being the respective angular displacements. If r_o is the maximum and ρ_o the minimum contact radii, the following relations hold:

$$p = \frac{2\,r_o\,\rho_o}{r_o + \rho_o}, \text{ and } e = \frac{r_o - \rho_o}{r_o + \rho_o} \qquad (W)$$

The distance between the centers of rotation O and O' of the ellipses is

$$q = r + r' = \rho_o + r_o = 2a,$$

where a is the semi-major axis.

From Equation (42) Part III, the angular displacements can be obtained for the above ellipses in this form

$$(1 - e)\tan\frac{1}{2}\theta' = (1 + e)\tan\frac{1}{2}\theta$$

which, by using Equation (W) and making substitutions, may be written as

$$\rho_o \tan\frac{1}{2}\theta' = r_o \tan\frac{1}{2}\theta \qquad (49)$$

By means of this result, the angular displacement of one rolling ellipse may be accurately computed in terms of the angular displacements of the other.

Referring again to Fig. 13 it is seen that

$$\text{arc } QA = \text{arc } QA'$$

these arcs are furthermore identical in shape. Hence if F is the free focus of L then

$$OA' = FA$$

$$\text{angle } QOA' = \text{angle } QFA = \theta'$$

and since by definition $FQ + r = 2a$, it follows that $FQ = r'$ as a consequence.

These results suggest a simple graphical method for obtaining the angular displacements, the contact radii and the instantaneous speed ratio of a pair of rolling ellipses. The method is as follows:

Draw an ellipse E, as in Fig. 14, using a convenient scale to suit the dimensions of one of the pair under consideration. From the focus F_1 draw a line to a point Q on the ellipse, OQ making a given angle θ with the major axis. This represents the angular displacement of one of the rolling ellipses. From the other focus F_2 draw a line to Q. The angle θ' that F_2Q makes with the major axis represents the angular displacement of the other rolling ellipse. The instantaneous contact radii will then be the ordinates F_1Q and F_2Q, respectively. The angular speeds of the rolling ellipses are inversely proportional to these radii, as was shown elsewhere.

It can be easily shown that the free foci F and F' in Fig. 13 will always keep the same distance $FF' = OO'$ from each other, and that the line FF' goes through the point of contact Q. Owing to the constancy of the distance between the free foci, they can be connected by means of a link without adding constraint to the motion of the ellipses. A pair of equal rolling ellipses with axes of rotation through the foci may therefore be reduced to a quadric chain with crossed links.

This is illustrated in Fig. 15, where the circles represent the paths of the free foci. The line FF' intersects the line of centers at Q, which corresponds to the point of contact of the equivalent rolling ellipses. The respective contact radii of these ellipses are then represented by OQ and $O'Q$. $OF = O'F' = r_o - \rho_o =$ the radii of the circles.

It was shown in Part II that in a pair of non-circular gears L and L', having, respectively, angular speeds ω and ω', and contact radii r and r', the relation

$$\omega'/\omega = r/r' \qquad (50)$$

holds. The division of Equation (47) by Equation (48) gives the speed ratio ω'/ω. The expression thus obtained, however, has the disadvantage of including angular displacements for both curves, one of which must be known in terms of the other by previous deter-

minations. It is better, therefore, to refer the speed to only one of the angular displacements.

Differentiating Equation (49) once with respect to time becomes

$$\rho_o \omega' \sec^2 \frac{1}{2}\theta = r_o \omega \sec^2 \frac{1}{2}\theta$$

which yields, after eliminating θ' by means of the same Equation (49)

$$= \omega' \frac{2R\omega}{R^2 + 1 - (R^2 - 1)\cos\theta} \tag{51}$$

where $R = r_o/\rho_o$. Equation (51) gives the angular speeds of L' in terms of the angular speed and angular displacements of L. If ω is constant, this equation is precisely the polar equation of an ellipse with ω' as radius vector, $2R\omega/(R^2 + 1)$ as the semi-latus rectum, and $(R^2 - 1)/(R^2 + 1)$ as the eccentricity. Hence, if the ellipse represented by Equation (51) is drawn, the values of ω' may be obtained graphically in terms of the angular displacements θ of L. When ω is constant, the maximum and minimum angular speeds of L' will be R and $1/R$, respectively. For this reason the constant R can be very appropriately termed the speed-ratio rating of a rolling ellipse.

Reversing the order of the variables in Equation (51) gives

$$\omega = \frac{2R\omega'}{R^2 + 1 + (R^2 - 1)\cos\theta'}$$

whence

$$\omega' = \frac{[R^2 + 1 + (R^2 - 1)\cos\theta']\,\omega}{2R} \tag{52}$$

which gives the angular speeds of L' in terms of its own angular displacements and the angular speed of the opposite ellipse L. Equation (52) will be found to be more useful in practice than Equation (51), especially when ω is constant.

The angle of obliquity of contact shown in Fig. 13 as ϕ is, by Equation (31, Part II) in the case of our present rolling ellipses, given by the following Equation

$$\tan\phi = -\frac{e\sin\theta}{1 - e\cos\theta} = -\frac{re\sin\theta}{\rho} \tag{53}$$

Referring to Fig. 13, the tangent TQ common to the two rolling ellipses through the point of contact Q goes through the point of intersection of the major axes of the ellipses. TQ bisects the angle OTO included by the major axes. The angle of obliquity ϕ may be obtained graphically from Fig. 14 as the angle that the normal line QN makes with F_1Q or F_2Q, this normal bisecting the angle F_2QF_1. Therefore,

$$\phi = \frac{1}{2}(\theta' - \theta) \tag{53'}$$

From Equation (32) Part II, tangential speed v is

$$v = \frac{r\omega}{\cos\phi} \tag{54}$$

The discussion of angular accelerations in this article will be limited to the case in which one of the rolling ellipses (usually the one representing the driving gear) rotates at constant angular speed. Let the angular speed of L be constant: ω = constant. Differentiating Equation (51) once with respect to time results in

$$\alpha' = -\frac{2R\omega^2(R^2 - 1)\sin\theta}{[(R^2 + 1) - (R^2 - 1)\cos\theta]^2}$$

or more simply

$$\alpha' = -\frac{\omega'^2}{2R}(R^2 - 1)\sin\theta \tag{55}$$

The angular accelerations α' of L' are seen here to be a function of the angular displacements of the driver L. To obtain the angular accelerations of L' in terms of this curve's own angular displacements, Equation (52) is differentiated once with respect to time, yielding the expression

$$\alpha' = -\omega\omega'\left(\frac{R^2 - 1}{2R}\right)\sin\theta'$$

Substituting the value of ω' given by Equation (42) in the above expression, and by a simple trigonometric transformation, the following is obtained

$$\alpha' = -\omega^2(A\sin\theta' + B\sin 2\theta') \tag{56}$$

where

$$A = \frac{R^4 - 1}{4R^2}, \text{ and } B = \frac{(R^2 - 1)^2}{8R^2}$$

Multiple Gearing-Linear Trains

Let L, L', and L'' in Fig. 16 be three engaging rolling ellipses with their respective centers of rotation O, O', and O'' in a straight line. After L has been displaced through an angle θ, L' and L'' will have been displaced through angles θ' and θ'', respectively, as is shown in the figure.

The relation of the angular displacements between L and L' is $\tan\frac{1}{2}\theta' = R\tan\frac{1}{2}\theta$, while the relation of the angular displacements between L' and L'' is $\tan\frac{1}{2}\theta'' = R\tan\frac{1}{2}\theta'$.

From these two relations is obtained

$$\tan\frac{1}{2}\theta'' = R^2\tan\frac{1}{2}\theta \tag{57}$$

as that between the angular displacements of the extreme curves L and L''.

The only formal difference between Equations (49) and (57) lies in the squaring of the constant R in the latter. Therefore, the angular kinematic relations between the extreme rolling ellipses above, whose maximum and minimum contact radii are in the ratio R are identical with the angular kinematic relations between the two members of a pair of rolling ellipses whose maximum and minimum radii are in the ratio R^2.

To generalize, consider a linear train of rolling ellipses having any number of members, with the speed-ratio rating of the curves being designated by R. Let the first ellipse be the driver of the entire train. Then the angular displacements of the nth driven ellipse with respect to those of the driver will be given by the expression

$$\tan\frac{1}{2}\theta_n = R^n\tan\frac{1}{2}\theta \tag{58}$$

where θ is the angular displacements of the driver and θ_n is the angular displacements of the nth driven ellipse. Obviously it follows that the equivalent single pair of rolling ellipses will have R^n as the speed-ratio rating.

Treating Equation (58) the same as Equation (49) was treated, the following is obtained for the case of linear trains for the nth driven ellipse.

$$\omega_n = \frac{2R^n\omega}{R^{2n} + 1 - (R^{2n} - 1)\cos\theta}$$

or

$$\omega_n = \frac{[R^{2n} + 1 + (R^{2n} - 1)\cos\theta_n]\,\omega}{2R^n} \tag{59}$$

The maximum and minimum speed ratios are seen to be here simply R^n and $1/R^n$, respectively, where ω is the angular speed of the driving ellipse and ω_n is the angu-

lar speed of the nth driven ellipse; and furthermore considering angular accelerations, if α is that of the driver and α_n that of the nth driven elipse, then

$$\alpha_n = - \frac{\omega_n^2 \quad (R^{2n} - 1) \sin \theta}{2 R^n}$$

or

$$\alpha_n = - \omega^2 (A_n \sin \theta_n + B_n \sin 2\theta_n) \qquad (60)$$

where

$$A_n = \frac{R^{4n} - 1}{4 R^{2n}}, \text{ and } B_n = \frac{(R^{2n} - 1)}{8 R^{2n}}$$

The angle of obliquity of contact between the nth and the $(n-1)$th driven ellipses is

$$\phi = \frac{1}{2} (\theta_{n-1} - \theta_n)$$

where θ_{n-1} is determined by

$$R \tan \frac{1}{2} \theta_{n-1} = \tan \frac{1}{2} \theta_n$$

The tangential speed between these ellipses is

$$v = \frac{r_n \omega_n}{\cos \varphi}$$

where r_n is the contact radius of the nth driven ellipse and is expressed by

$$r_n = \frac{p}{1 + e \cos \theta_n}$$

The graphical method described in connection with Fig. 14 may be extended to cover linear trains. Fig. 17 illustrates the method applied to the treatment of four rolling ellipses L_o, L_1, L_2, and L_3 represented by the ellipse E drawn to a suitable scale. The segment $F_1 Q'$, representing the contact radius of L_o, is drawn at an angle θ_o with the line through the two foci F_1 and F_2, this angle being the angular displacement of L_o. The segment F_2Q' and the angle θ_1 will be the contact radius and the angular displacement of L_1. The line F_2Q' is produced to meet E at Q'', and the segment F_2Q'' will be the complementary contact radius of L_1. The segment F_1Q'' and the angle θ_2 will be the contact radius and the angular displacement of L_2, and F_1Q''' its complementary contact radius. The contact radius and angular displacement of L_3 will, finally, be the segment F_2Q''' and the angle θ_3.

Balancing of Elliptical Gear Drives

As might be expected, the inertia torque (the so-called "flywheel effect") developed by the irregular motion of elliptical gear drives is considerable, being in the majority of cases the greatest part of the load carried by the teeth. The moving parts affected by the motion of an elliptical gear drive, therefore, should be as light as possible. If the actual load carried by the drive is concentrated in a small interval of the cycle of motion, as is the case in shearing machines, it will be found desirable to introduce a flywheel to absorb the shock and to steady the drive. But the introduction of such a flywheel will necessarily increase the inertia torque. Hence, in this case the only possible way of reducing the load on the teeth lies in the use of a device to balance out the inertia torque.

Let L_n, in Fig. 18, be the last gear of a linear train composed of n driven elliptical gears, and one driving elliptical gear of constant angular speed ω. Let I be the angular moment of inertia of all the masses rotating with the shaft of L_n. By Equation (60) the inertia torque will in this case be

$$T_i = \alpha_n I = - \omega^2 I A \sin \theta_n - \omega^2 I B \sin 2\theta_n \qquad (61)$$

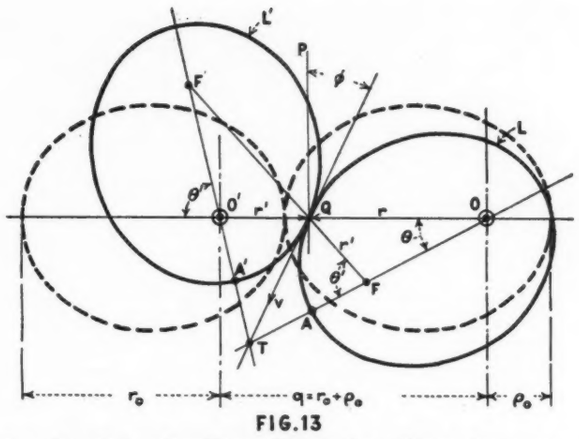


FIG.13

Fig. 13—Pair of rolling ellipses for illustrating contact radii and angular displacement relations

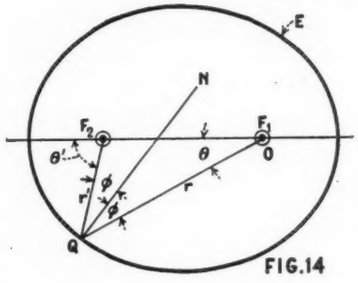


FIG.14

Fig. 14 — Simple graphical method for obtaining angular displacements, contact radii and instantaneous speed ratios

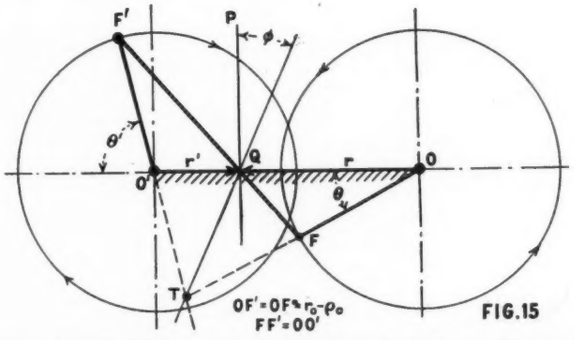


FIG.15

Fig. 15—Quadric chain with crossed links is kinematically equivalent to a pair of rolling ellipses with axes of rotation through the foci

To balance out this inertia torque completely, the spring arrangement shown in Fig. 18 must produce a torque T_s equal and opposite to the inertia torque, so that

$$T_i + T_s = 0 \qquad (62)$$

At the end of the shaft carrying L_n is attached a crank OP of length a, disposed at right angles to the major axis of L_n, as shown. A long compression spring S_1 is then attached to the end P of the crank and is pivoted at a fixed point A_1 on a line through O perpendicular to the line of centers of the gear train (assuming for simplicity, that all the elements in Fig. 18 are in a single plane). Similarly another long spring S_2 is attached at P and pivoted on a fixed point A_2 on the line of centers of the train.

Clearly, spring S_1 could be a tension spring if it was rotated 180 deg. in the plane of the figure, or if the crank was rotated the same amount with respect to L_n and then held fast on the shaft. It is not necessary that the springs be directly attached to the crank, a system

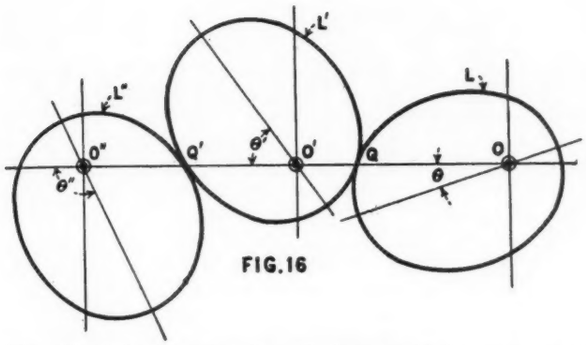


FIG.16

Fig. 16—Showing angular displacements in a train of elliptical rolling curves

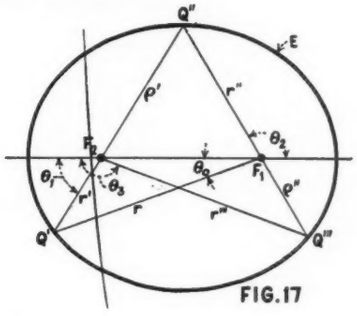


FIG.17

Fig. 17 — Graphical method extended to cover linear train

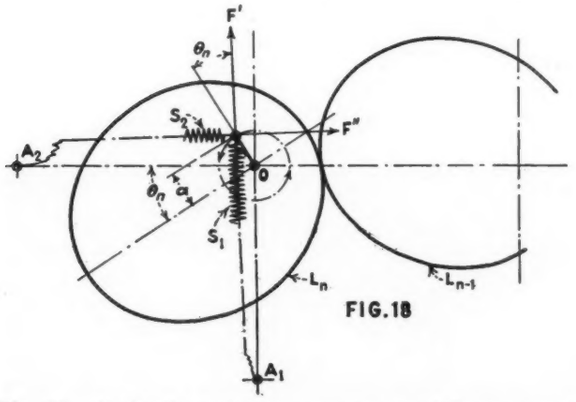


FIG.18

Fig. 18—A simple spring arrangement to balance inertia torque developed by the irregular motion of elliptical gear drives

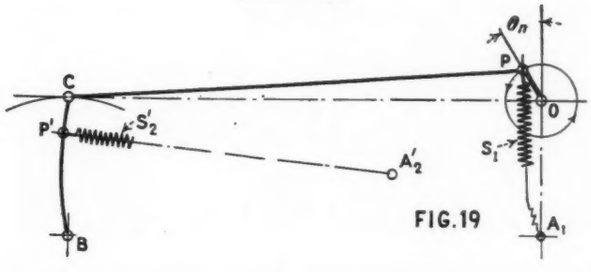


FIG.19

Fig 19—Scheme for varying spring gradient to suit the speed of the machine while the machine is in motion

of levers could be used whereby short springs could be employed that would produce effects the same as directly attached long springs.

The springs S_1 and S_2 are required to be long so that their forces F' and F'' on the crank may be considered constant in direction.

Force F' of spring S_1 produces a torque

$$T_n' = a\,F' \sin \theta_n$$

But

$$F' = F_1 - a\,K_1 \cos \quad _n$$

where F_1 is the mean force of S_1, and K_1 is the spring constant or gradient of S_1. Hence

$$T_n' = a\,F_1 \sin \theta_n - a^2\,K_1 \sin \theta_n \cos \theta_n$$

$$= a\,F_1 \sin \theta_n - \frac{1}{2} a^2\,K_1 \sin 2\,\theta_n$$

The spring S_2 works alternately in tension and compression, its mean force being zero. Hence, its force is simply

$$F'' = a\,K_2 \sin \theta_n$$

producing the torque

$$T_n'' = a^2\,K_2 \sin \theta_n \cos \theta_n = \frac{1}{2} a^2\,K_2 \sin 2\theta_n$$

where K_2 is the gradient of S_2.

The total torque produced by the two springs is

$$T_n = T_n' + T_n'' = a\,F_1 \sin \theta_n + \frac{1}{2} a^2\,(K_2 - K_1) \sin 2\theta_n \quad (63$$

Substituting now the values given by Equations (61) and (63) in (62), the following equation is obtained

$$\left[aF_1 - \omega^2 I A_n\right] \sin \theta_n + \left[\frac{1}{2} a^2\,(K_2 - K_1) - \omega^2 I B_n\right] \sin 2\theta_n = 0$$

The functions $\sin \theta_n$ and $\sin 2\theta_n$ being linearly independent here, to satisfy the equation it is necessary that each algebraic sum of the terms inclosed in each of the brackets be equal to zero. From this condition, therefore

$$F_1 = \frac{\omega^2\,I\,A_n}{a} \quad (64)$$

$$K_2 - K_1 = \frac{2\,\omega^2\,I\,B_n}{a^2} \quad (65)$$

which relations allow the calculation of the springs from the constants of the drive. The second member of Equation (65) is positive, and hence the gradient K_2 of S_2 must be steeper than the gradient K_1 of S_1. Therefore, K_2 could never be zero, for then S_1 would have a negative gradient, which no single or coaxially compounded spring is known to possess. In fact, this was the reason for introducing a second spring S_2 at right angles to S_1.

The units implied in the relations expressed in Equations (64) and (65) are absolute, ω being in radians per absolute unit of time. Thus, in the British system of absolute units, a is in ft., I is in pounds-mass-ft.^{2}, F_1 is in poundals, K_1 and K_2 are in poundals per foot, and ω is in radians per second. A_n and B_n have no dimensions. If engineering units are used, the proper constants should be introduced in these relations.

To satisfy Equation (65), means should be employed to vary K_2. The scheme shown in Fig. 19 will accomplish this. A spring S_2' of zero mean force is pivoted at a fixed point A_2' and attached to a movable point P' on the lever BC, the latter being pivoted at a fixed point B. A long connecting rod CP is used to transmit the force of S_2' to the crank OP. The gradient of this system reduced to that of the equivalent directly acting spring along CP, considering C fixed, would clearly be expressed by

$$K_2 = \frac{BP'}{BC} K_2' \text{ approx.}$$

where K_2' is the gradient of S_2'. By moving P' along BC, K_2 may be varied at pleasure. Points P' and A_1 could be so interlocked by means of levers or cams that they could be moved simultaneously while the machine is in motion.

QUESTION AND COMMENT

Modifying Geometric Progressions in Speeds

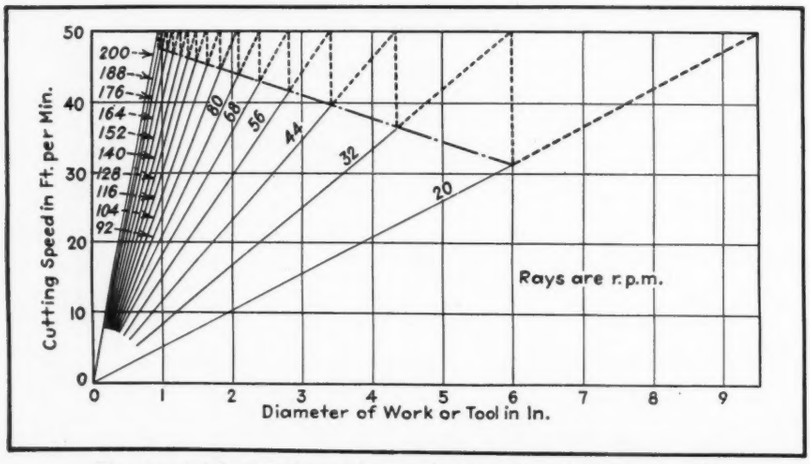


Fig. 1—Arithmetical speed step distribution. Speed range 20 to 200 r.p.m. Step increment is 12 or 16 speeds

To the Editor:

● In the December 1936 issue of *Product Engineering,* E. J. Rantsch, suggests a modification of the geometric progressions of speeds so that at higher speeds or at smaller diameter of work or tool the successive spindle speeds will be in smaller increments. Tables and a diagram supporting his suggestion are shown. He maintains that in geometrical progression the difference from step to step in the lower speeds is too small while between the successive higher speeds the steps are too large.

However, actually such a grouping of speeds in machine tools is not desired. To the contrary a grouping of steps closer at the lower speeds and further apart at the higher speeds is far more desirable. It should be kept in mind that revolutions alone are not of primary importance; Diameters and cutting speeds are the basis of speed calculations. If it were not so, we probably would still have arithmetical progression as the basis for figuring the series of speed changes required for a machine tool. In Fig. 1 is shown graphically the relation between cutting speed and work or tool diameter and how the steps are distributed over a given speed range, if based on arithmetical progression. The dot and dash line at the lower part of the saw-tooth diagram shows a uniformly increasing difference in cutting speeds.

Mr. Rantsch's figures of suggested speeds as given in Table II in his article are represented in the irregular saw-tooth diagram shown in Fig. 2. It can readily be seen that a closer range of steps at higher speeds would be more easily accomplished by arithmetical progression.

The saw-tooth diagram in Fig. 3 ilustrates speed step distribution with the geometrical progression method as basis. The advantage of such an arrangement is evident because:

1. The speed drop is constant between each successive step as seen by the dot and dash line.

2. Steps are better distributed, for they are closer at low speeds and further apart at high speeds.

Many machine tools today are geared on the basis of geometrical progression for desired spindle speeds, the only difference existing between similar types is the wide variation in the geometrical progression ratios

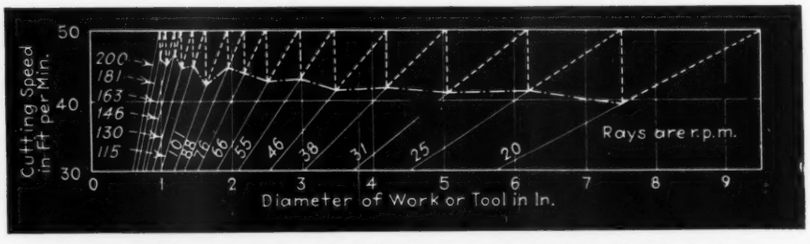


Fig. 2—Modified speed step distribution as suggested by Mr. Rantsch. Speed range 20 to 200 r.p.m. Speed drop is irregular

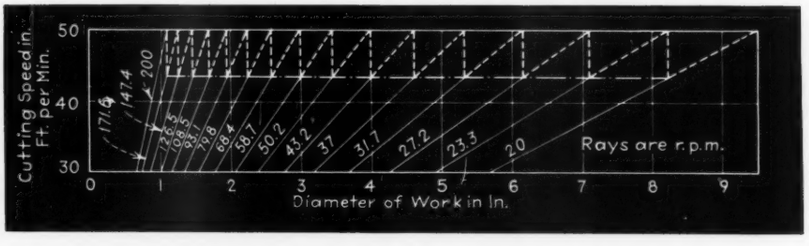


Fig. 3—Geometrical speed step distribution. Speed range 20 to 200 r.p.m. with 1.165 geometrical ratio for 16 speeds

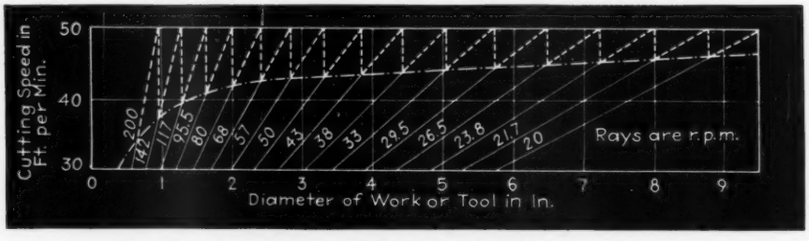


Fig. 4—Logarithmical speed step distribution in a speed range of 20 to 200 r.p.m. having 16 speeds

used. Machine tools of one kind require a certain ratio and those of another kind require a different ratio, depending mostly on the size of work to be machined. Two milling machines designed for the same purpose but built by two different manufacturers may show a considerable difference in the ratios used for distributing the speed steps over a certain speed range.

In Fig. 4, is shown a further attempt to distribute the steps more evenly over the whole range of speeds, founded on a logarithmic basis. The diagram given is only approximated. It would take too much space to give the derivation here. As much as such an arrangement would be desired it has found little application, if any, and is only mentioned to show a further possibility of shifting the steps to the more desired end according to a certain law.

With the pending standardization of speeds any arrangement differing from the one based on geometric progression seems to be out of question. The proposed standard is based on a decimal geometric series coinciding with the series of preferred numbers. A standard on this basis is in effect in Germany and is also the basis for an international standard. The American standard can be tied to this international standard very nicely with a slight shift in its basic starting number necessitated by the different motor-speeds common in this country. A detailed discussion of this is given in the articles "Geometric Progressions in Machine Tool Speeds," J. Decker, *American Machinist*, Vol. 74, No. 19, 1931 and "Significance of Speed Standardization" by J. Decker and Acres, *Mechanical Engineering*, Vol. 55, No. 4, 1933.

In view of the coming standardization on the geometrical progression basis there seems to be very little need for a deviation from such a standard once it is adopted. In fact any other step distribution would make it impossible to take advantage of such a standardization which expresses itself not only in simplified design but also in simpler calculations and the setting of standards in production work.

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RCA Manufacturing Company, Inc.
Camden, N. J.

To the Editor:

● Having experienced that disillusionment which cannot fail to proceed from any attempt to embody series other than the geometric in speed ranges, I make no apology for this brief comment on the proposal of Mr. E. J. Rantsch, in the December, 1936, number.

It is not often realized that most of the usual gear arrangements forbid any regular speed progression; that is, rise or fall in a curve obeying a uniform law throughout. An analysis of your contributor's suggested modification of the geometric range will illustrate my meaning.

In most speed-changing mechanism, several series of speeds are obtained through each pair of gears. Thus a 16-speed range might be obtained by three shafts, a number of gears on the primary shaft driving a like number of gears on the secondary shaft (Intermediate gears) and another or the same number of gears on this shaft driving a like number of gears on the final shaft. Thus, 4 pairs of gears might drive 4 pairs; 8 pairs, 2 pairs; 2 pairs, 8 pairs. Respectively these might be termed 4/4, 8/2, 2/8 arrangements.

Consideration of the 4/4 arrangement will illustrate the fault it is desired to exhibit. Let the pairs of gears between one shaft and the intermediate be denoted by *A*, *B*, *C*, *D*, and the pairs between the other shaft and the intermediate by 1, 2, 3, 4. Then, according to your contributor's Table II, the speeds each combination of these should give is:

	A	B	C	D
1	20	46	88	146
2	25	55	101	163
3	31	65	115	181
4	38	76	130	200

A little thought will reveal that, within the limits imposed by integral numbers of teeth, it is always possible to select gears to give speeds *A*1, *A*2, *A*3, *A*4, *B*1, *C*1, *D*1. But, in getting these speeds, all the gears have been fixed and hence the remaining speeds are definitely fixed. Little study is needed to show that the speeds in each column should be in the same ratios; that is, as 20 : 25 : 31 : 38. So the preceding table must be rewritten as:

	A	B	C	D
1	20	46	88	146
2	25	57	110	182
3	31	71	136	226
4	38	87	167	277

The error will best be seen by comparing the differences. We note, for instance, a rise of but 1 between *B*4 and *C*1, whereas between *C*4 and *D*1 there is a fall of 21 instead of a rise. The other gear combinations would have revealed similarly erratic differences.

The value of certain modifications is, despite our criticism, not to be denied. One example within ken was the provision of some extra high and one extra low speed for contingencies, the speeds for general work being still "geometrical." But even these were not obtained without additioned gears.

—FRANCIS W. SHAW
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Charts for Circular Segments

L. D. HAGENBOOK
De Pere, Wisconsin

● The accompanying chart for solving the properties of circular segments has proven useful in my office, because of the time required in making such calculations. The chart gives in curve form, solutions of the area, section modulus, moment of inertia and distance to fiber from the neutral axis for circular segments of various heights. The curves shown at the left in the chart are similar to those shown at the right of the chart except that they are plotted to a larger scale. In using the curves, it is necessary to

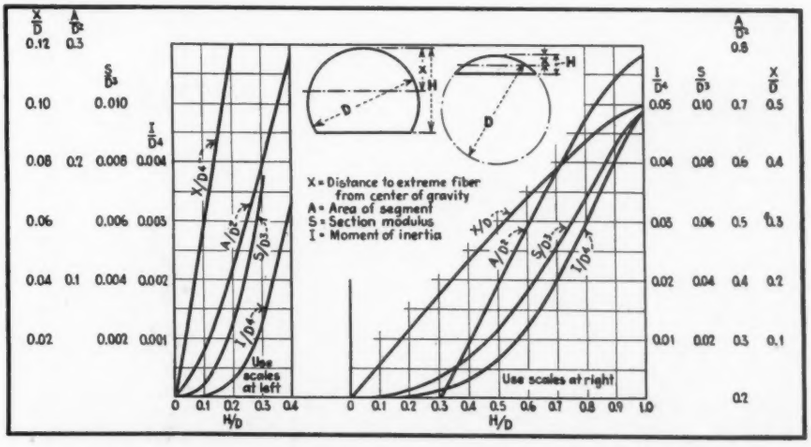

X = Distance to extreme fiber from center of gravity
A = Area of segment
S = Section modulus
I = Moment of inertia
Use scales at left
Use scales at right
H/D

Chart for calculating the area, section modulus, moment of inertia and distance to furthest fibers in the segment of a circle

reduce all values to proportions of a 1 in. diameter circle. The height of the segment is represented by H/D (height of segment divided by diameter of circle) and forms the common abscissa. Values may be read from one of the vertical scales graduated for its corresponding curve. These values are for a 1 in. diameter circle. For a circle of any other diameter simply multiply by the diameter raised to the power indicated; that is for I, multiply by D^4; for S, multiply by D^3; for A, multiply by D^2; for X, multiply by D. The accompanying examples show how the chart is used:

EXAMPLE 1: Find I for a segment $2\frac{1}{2}$ in. altitude, lying in a circle 4 in. in diameter.

SOLUTION: $H/D = 0.625$. From the curve $I/D^4 = 0.0142$.
$I = D^4 \times 0.0142 = 256 \times 0.0142 = 3.6352$.

EXAMPLE 2: The diameter of a segment is 2 in. and the area is to be 1 sq. in. Find the height.

SOLUTION: Curve A/D^2 applies in this case. $A/D^2 = 1/4 = 0.25$
0.25 corresponds to $H/D = 0.355$ $H = 0.355 \times 2 = 0.710$ in.

Exact formulas for the properties are as follows:

In these formulas let
$r = D/2$ and $y = r - H$

Then $A = y\sqrt{r^2 - y^2} + r^2 \sin^{-1}(y/r)$

$x = r\ (2/3\ A)\ (r^2 - y^2)^{3/2}$

$I = (2y^2 - r^2/4)\ y\ (\sqrt{r^2 - y^2} + [(r\ 4/4) \sin^{-1}(y/r)] - x^2 A$

$S = I/x$

Drop-Forging Design

● When parts are to be made by drop-forging, the design and shape of the piece must be such as to lend itself to formation by plastic deformation of hot bar material between two dies. As explained by the engineers of the J. H. Williams Company, the designer must know where the "parting line" is to be, as it is from the parting line, that all drafts are measured. Although many pieces are forged in dies having a parting line in one plane, the parting line of the lever shown in the illustration must be in several planes so that the dies will part both on the centers of the $\frac{7}{16}$-in. diameter hubs and also on the bosses having $\frac{7}{16}$-in. radii. Such dies are called "locked dies."

The "forging plane" which is an imaginary plane perpendicular to the line of reciprocation of the dies, must also be decided. In the illustration the forging plane is the same as the two parallel portions of the parting line, but in some instances, to take advantage of the natural draft, work can be forged in a tipped position in the dies. To avoid the possibility of hooking, undercutting, or interlockng, whereby the dies are prevented from separating, a full-sized forging drawing, such as shown in the illustration, is made. This drawing contains only the dimensions and other information necessary to make the forging.

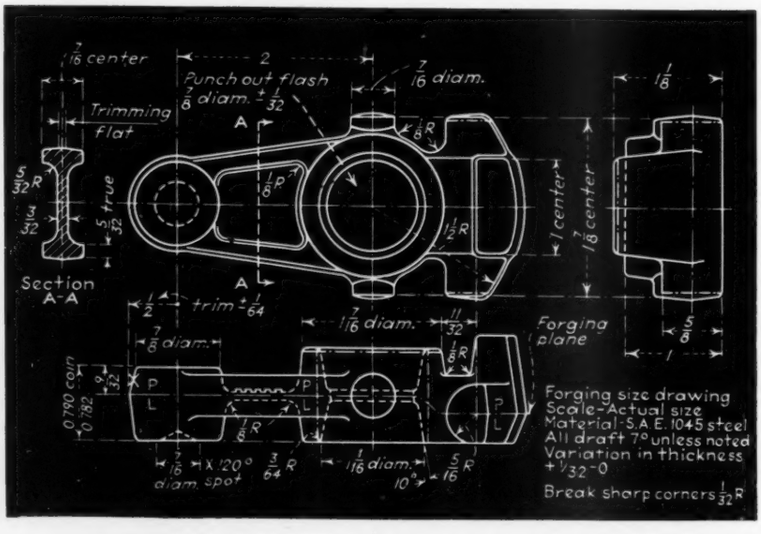
Although drop forgings can be made of any desired analysis of carbon, alloy or stainless steel, iron, tobin bronze, monel metal, aluminum or beryllium copper, detailed chemical and physical characteristics should be specified. Machinability requirements, whether forgings are to be delivered normalized or heat-treated, hardness or physical properties, grain flow or grain size should also be given when possible.

Usually the standard draw, clearance or draft is 7 deg. for all exterior surfaces. Lesser exterior draft angles such as 3 and 5 deg. are frequently permissible, but always at increased cost because of more rapid die failure. Interior draft such as for holes in the forging which correspond to frustrums of cones in the dies, have a 10 deg. draft. When a regular detailed drawing is submitted to the drop-forging manufacturer, he always adds draft to any surfaces that are perpendicular to the forging plane. In the table are given die-draft equivalents for various draft angles and depths. Because draft angles are subject to increase as the dies wear, it is best to show all dimensions to points deepest in the die and not overall or to the parting line. If an overall dimension is of importance, it should be emphasized by suitable descriptive notes. Length of forgings may vary as much as ±0.003 in. per inch of length.

Since sharp corners on the forging tend to cause die checks or cracks, and thereby premature die failure, projecting corners or edges are usually broken with $\frac{3}{32}$ to $\frac{1}{8}$ in. radii. Fillets should always be as large as possible to assist flow of hot metal. Dimensions for thickness of ribs and diameters of bosses having corner radii are always specified tangent to the radii. When both ends of a boss or hub are the same diameter, the draft angle of the shorter end is made greater in the shallow die to coincide with the longer side at the parting line. An example of this is shown at X in the illustration. A dimension such as given in the drawing of $\frac{1}{2}$ in.-trim $\pm \frac{1}{64}$ in., controls overall length to some extent, but ordinarily this trimming is not held to limits unless specified.

In the forging operation, a surplus or thin fin of metal known as flash remains around the outline of the piece at the parting line. This flash is trimmed off close to the body by a hot or cold operation, leaving a trimming flat of $\frac{3}{32}$ to $\frac{1}{4}$ in. in width, this width depending on the size of work and newness of the dies. Variations in trimming may cause a slight bulge of flash, but a dimension and note as given above will overcome this objection.

Narrow or deep depressions in the forging should be avoided whenever possible as the male parts of the dies which form them are weak and do not withstand the heat and pressure of the forging operation. When absolutely necessary, a minimum of 10 deg.-draft and large fillets are used. The depth of the depression or forged hole in either half of the die should not exceed $\frac{2}{3}$ of the smallest width of the depression. Thin ribs or webs are also to be avoided especially in alloy steel forgings, thicknesses of $\frac{3}{32}$ or $\frac{1}{8}$ in. and greater being preferable for best

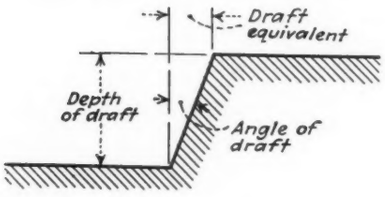


Typical detailed drawing for a part to be forged, showing parting lines, drafts, tolerances and notations

production and lowest cost. All thickness dimensions perpendicular to the forging plane may vary from 0.015 in. on small work to $\frac{3}{16}$ in. on large.

Finish lines for machining are shown on the drawing by dot and dash lines. Usually $\frac{1}{32}$ to $\frac{1}{16}$ in. is left on finished surfaces of medium size forgings. On small pieces, the allowance may be as little as $\frac{1}{64}$ in., and $\frac{1}{8}$ in. on large forgings. If a particular finish surface also has draft, the draft is considered as additional thereto. On surfaces where it is desirable to eliminate machining or grinding operations, but where close tolerances are required, coin-pressing of small and medium size forgings can be specified. In this type of production, tolerances as close as 0.005 in. can be held. For bosses or hubs having finished holes, drill spots of 105- to 120-deg. included angle can be provided on surfaces approximately parallel to the forging plane. Raised letters, such as trade marks, part numbers and other markings can be placed on any suitable areas which approximately parallel the forging plane.

DIE-DRAFT EQUIVALENTS

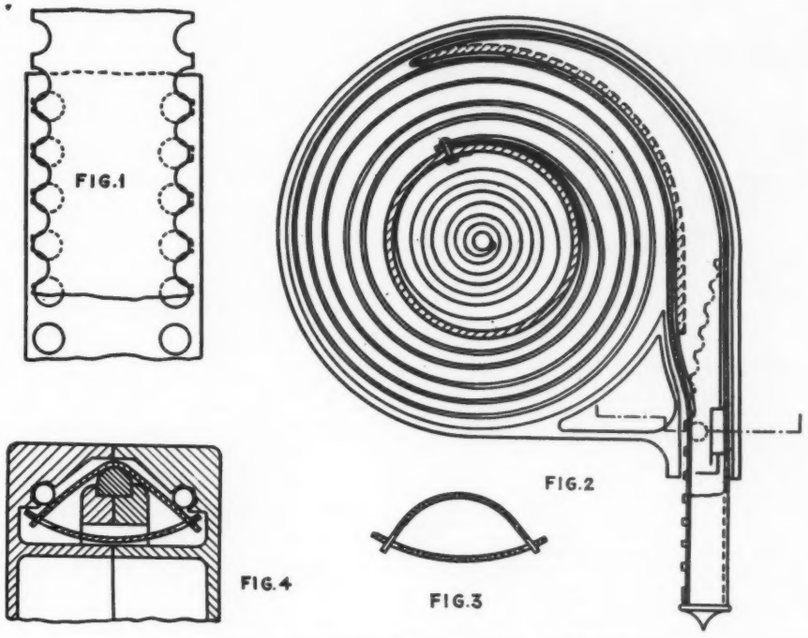


Depth in In.	Angle of Draft		
	5 deg.	7 deg.	10 deg.
1/32	0.0027	0.0038	0.0055
1/16	0.0055	0.0077	0.0110
3/32	0.0082	0.0115	0.0165
1/8	0.0109	0.0153	0.0220
5/32	0.0137	0.0192	0.0276
3/16	0.0164	0.0230	0.0331
7/32	0.0192	0.0269	0.0386
1/4	0.0219	0.0307	0.0441
9/32	0.0246	0.0345	0.0496
5/16	0.0273	0.0384	0.0551
11/32	0.0301	0.0422	0.0606
3/8	0.0328	0.0460	0.0661
13/32	0.0355	0.0499	0.0716
7/16	0.0383	0.0537	0.0771
15/32	0.0410	0.0576	0.0827
1/2	0.0438	0.0614	0.0882
9/16	0.0492	0.0691	0.0992
5/8	0.0547	0.0767	0.1102
11/16	0.0601	0.0844	0.1212
3/4	0.0656	0.0921	0.1322
13/16	0.0711	0.0998	0.1433
7/8	0.0766	0.1074	0.1543
1	0.0875	0.1228	0.1763

Errata

• On page 11 of the January, 1937 issue in the article "Types of Insulated Conductors for Electrical Apparatus," the Equation $D = 1.56 \sqrt{N}$ should read $D = 1.156 \sqrt{N}$. In the example immediately following the equation is given correctly.

Collapsible Tube That Winds on a Reel



Figs. 1 to 4—Construction details of a collapsible tube and the reel on which it is wound

• A two-piece stiff hollow metal rod that can be wound on a reel like a steel tape rule has been developed in Germany and patented in that country and in United States. As shown in the illustration, Fig. 1, the rod consists of two spring steel bands one of which is provided with a serrated edge while the other band has perforations along each edge. In the collapsed state, as shown in Fig. 2, the two bands are wound on a reel lying flat on one another.

When the bands are drawn out of the casing, the serrated member bends so that its cross-section takes a shape approximately a 90 deg. arc, the tips of the serrations fitting into the holes of the perforated member which also takes a curved shape, as shown in Fig. 3. In this way the two interlocked bands form a hollow rod of considerable stiffness. In Fig. 4 is shown the ball guide which helps to interlock the two members as they are pulled out of the case.

This type of collapsible rod has been used for adjustable lamp bases, in dental equipment, adjustable short-wave antenna for airplane and radio cars, and adjustable stands for show-window display.

—F. LOEWENBERG
New York, N. Y.

Can You Work This One?

H. E. SMITH
Newark, N. J.

This month's problem—

Bottle in the Stream

A man rowing upstream passes a bottle floating down stream at the same time that he passes a certain tree on the bank of the stream. He continues rowing steadily upstream for 30 min., whereupon, deciding that there might be something worthwhile in the bottle, he turns around and rows downstream after it at the same speed relative to the water. When he finally catches up with the bottle, it is two miles below the tree where he first saw it. How fast was the stream flowing?

Solution to Last Month's Problem—

Rope, Pulley and Weight

The weight and the sailor will go up together until they are stopped in their rise when they reach the pulley. It may be hard to visualize because the factors concerned are imaginative, namely a frictionless pulley and a weightless rope. However, the problem is based on the principle of equal and opposite reaction.

Methods of Sealing

Seals as applied to plain or sleeve-type bearings for retaining the lubricant or for excluding foreign materials or a combination of both

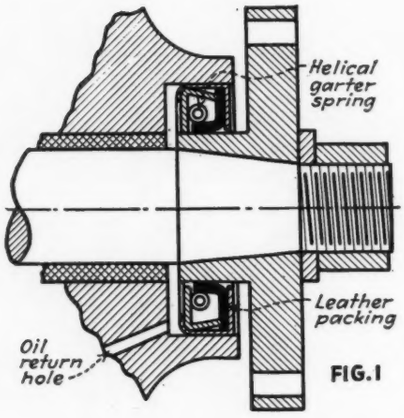


FIG.1

Fig. 1—For retaining lubricant, the seal is assembled with the flanged leather projecting toward the bearing. The leather packing is clamped near the outer edge of the flange by the inner of two telescoping metal cups, thereby assuring a tight joint at the face. A garter type spring compresses the leather about the shaft. Should misalignment occur the seal is maintained by virtue of the flexibility of the leather and garter spring. To drain off the surplus oil passing the end of the bearing, a small hole is drilled in the casting connecting the reservoir.

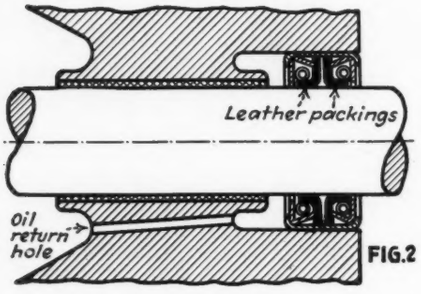


FIG.2

Fig. 2—Installation of double seal unit for retaining lubricant in bearing recess and for guarding against entrance of foreign material. Seal is of same general construction as shown in Fig. 1 except that two flanged leathers are mounted opposed to each other.

Fig. 3—Used for the same general purposes as arrangement shown in Fig. 2. Seal has but one garter spring for the oil retention leather flange. The leather washer for dust exclusion shown at right has a beveled lip which contacts the shaft.

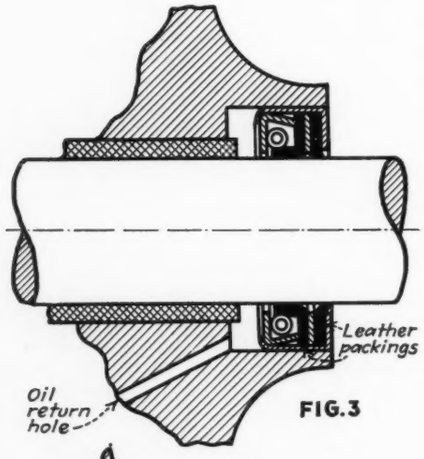


FIG.3

Fig. 4—Sometimes felt is used on the dust exclusion side of the seal in place of leather shown in Figs. 2 and 3. Both sealing materials are retained by spinning the outer casing over the leather clamping cup.

Fig. 5—Where there is considerable difference in the diameters, the face of the shoulder thus formed can be utilized as the sealing surface. A soft ring of cork or leather is beveled at the outer surface as shown. A flat spiral spring, coiled to a greater diameter than the hole in the sealing material expands the packing outwards against the beveled ring and wedges it against the face of the shaft shoulder.

Fig. 6—Working on the same wedging principle as that shown in Fig. 5, except that the packing is beveled on the inner surface and is retained by a sheet metal flange. The cork or leather sealing material is compressed against the two bearing surfaces by a garter spring as shown. Seals shown in Figs. 5 and 6 are limited to approximately 1/32 in. end play.

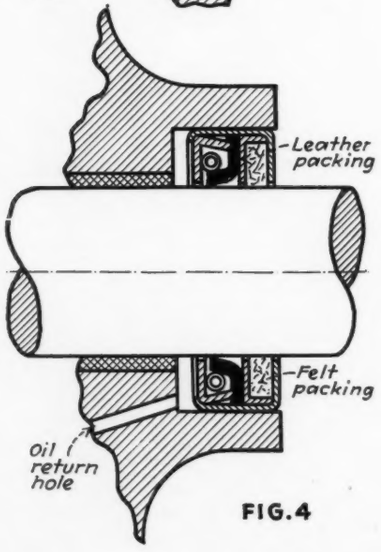


FIG.4

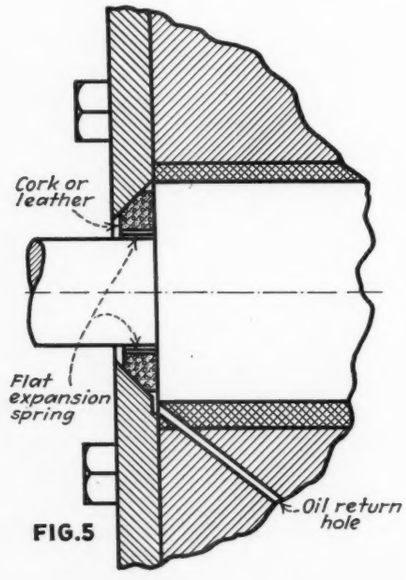


FIG.5

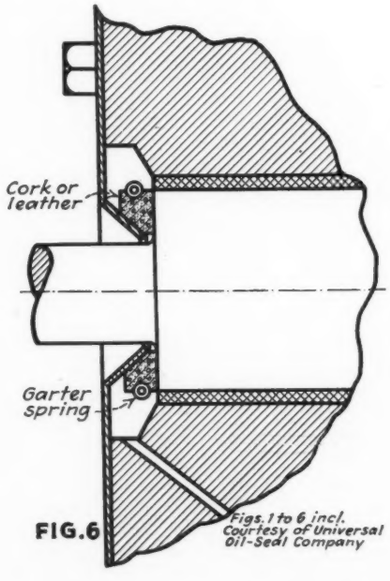
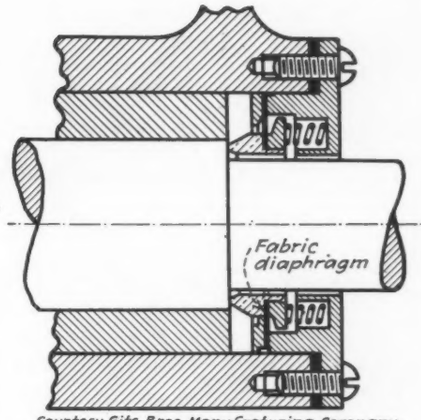


FIG.6

Figs. 1 to 6 incl. Courtesy of Universal Oil-Seal Company

Sleeve Bearings

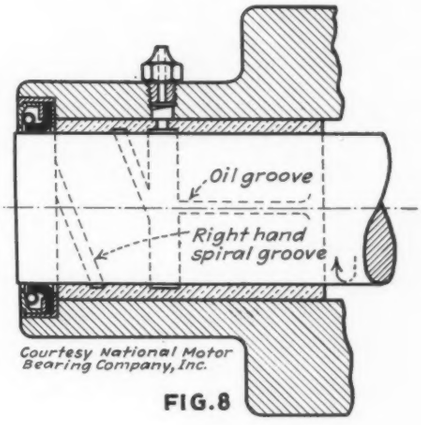
R. G. N. EVANS
The Bunting Brass & Bronze Company



Courtesy Gits Bros. Manufacturing Company

FIG. 7

Fig. 7—Another type of seal wherein a bronze ring bears against the shoulder of the shaft. The sealing materials is in the form of a diaphragm of heat resisting fabric which retains oil in the bearing and excludes dirt. In the flanged member that is screwed to the housing is a series of compression springs which hold the ring against the shaft shoulder. These springs not only take up wear but provide for end play of the shaft. To avoid torsional strain on the diaphragm guide pins are used between the outer flange and spring bearing washer.



Courtesy National Motor Bearing Company, Inc.

FIG. 8

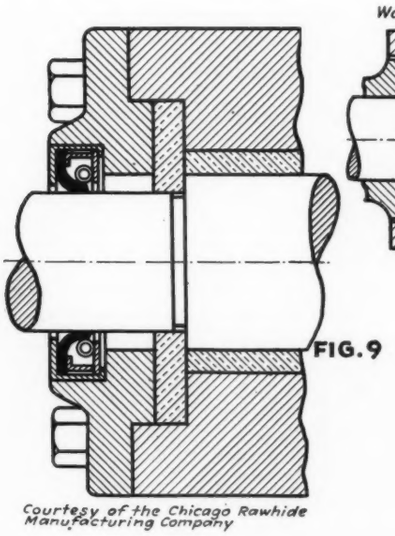
Fig. 8—When grease is used as a lubricant, it is sometimes desirable to assemble a single seal to keep dirt from reaching the bearing rather than retain the grease in the bearing. The illustration shows an installation wherein a right hand spiral groove is cut in the bearing bore to lead the lubricant outwards. Surplus grease is forced past the seal thereby keeping the bearing clean.

Fig. 9—Leather flange seal with garter spring mounted in a flanged end plate. Spring tension is such as to give small area of contact between leather and shaft, thereby minimizing friction. Bronze thrust washer between bearing and bearing housing.

Fig. 10—The labyrinth seal shown does not rely on non-metallic materials but on the small clearances within the assembly. A steel washer contacting a bronze thrust washer, is clamped against the shaft shoulder after the formed dust seal cup is pressed into the counterbored hole.

Fig. 11—When oil seals are to be installed after a mechanism has been assembled or to preclude the necessity of disassembling heavy shafts and bearings when making seal renewals, split seals can be used in such installations. The spreader spring and packing ring are split while the retaining cup is made in two halves. The packing is scarf-cut to form an oil-tight joint when assembled.

Fig. 12—Another mounting of small worm drive shaft for domestic washing machine and domestic stoker. The composition sealing material is held against the shaft by a V-formed spreader spring having serrated edges which nests into the sealing ring. The angle of the V in the spring is greater than the groove in the seal so that the fingers of the spring exert a light pressure on the sealing lip. An oil return hole is drilled outside the bearing to relieve built-up pressure against the seal.



Courtesy of the Chicago Rawhide Manufacturing Company

FIG. 9

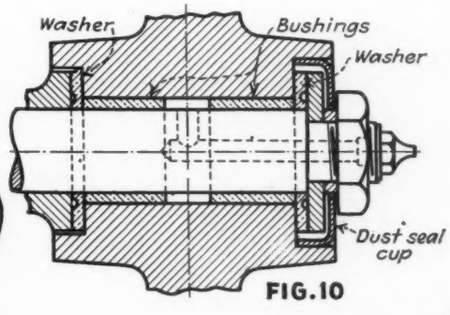
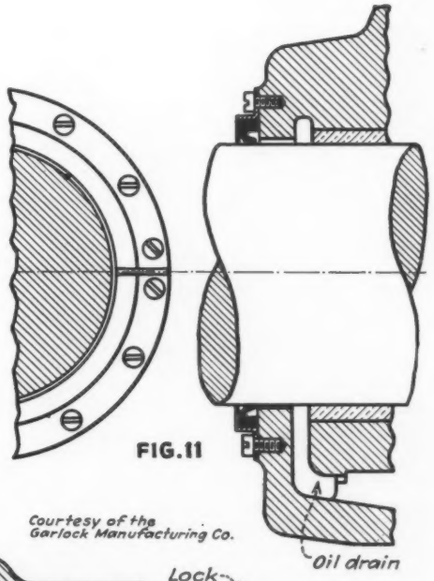


FIG. 10



Courtesy of the Garlock Manufacturing Co.

FIG. 11

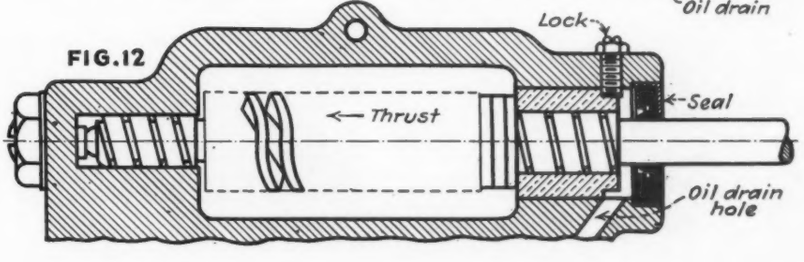


FIG. 12

NEWS

Lincoln Foundation Announces Prize Contest

TO STIMULATE an intensive study of arc welding and arc-welded designs, $200,000 in prizes will be distributed by the James F. Lincoln Arc Welding Foundation among 446 separate prize winners for papers dealing with arc welding as a primary process of manufacture, fabrication, construction and design. The principal prize winner will receive not less than $13,700, and prizes ranging from $7,500 to $100 will be awarded to each of 178 contestants whose papers are adjudged worthy of honorable mention.

In order to assure equal competitive opportunity, similar prizes are offered in each of eleven major divisions of industry covered by the contest. These divisions are: automotive, aircraft, railroad, watercraft, structural, furniture and fixtures, commercial welding, containers, welderies, functional machinery and industrial machinery. Wide diversification of awards will be effected by further dividing each major industry into various sub-classifications, with entrants required to select in advance the particular sub-classification for which their papers are submitted.

When accepted by the Jury of Awards as being properly classified, each paper will be in competition in its particular sub classification for five initial prizes established for that group. These prizes are $700, $500, $300, $200 and $150, respectively. From among these sub-classification winners, four papers will be selected in each major industry to receive additional prizes of $3,000, $2,000, $1,000 and $800. In addition, the semi-final winners in the various divisions will be considered as possible recipients of the four main prizes. These range from $10,000 to $3,500, with the winner of the grand prize receiving not less than $13,700 for his paper.

In the automotive field, the papers submitted will be divided under the four sub-classifications of engines, bodies, frames and trailers. In this field there will be a total of 24 prizes.

In the automotive field there will be 14 prizes in two sub-classifications, namely, Engines and Fusilage.

In the railroad field there will be 24 prizes in four sub-classifications —locomotives, freight cars, passenger cars and locomotive and car parts.

The watercraft industry will have two sub-classifications; commercial craft and pleasure craft, for which there will be a total of fourteen prizes.

Twenty-four prizes will be available in the structural field which is divided into the four sub-classifications of buildings, bridges, houses, and miscellaneous.

In the furniture and fixtures division, a total of fourteen prizes will be divided between the two sub-classifications which are house furniture and office furniture.

For the commercial welding division, there will be a total of fourteen prizes in the two sub-classifications, namely, job shops and garages.

There will also be a total of fourteen prizes in the containers division, which division is split into two sub-classifications, namely, contents stationary and contents moving.

In the division of industry there will be fourteen prizes distributed in two sub-classifications, namely, commercial welderies, and welderies that are departments of manufacturing plants.

In the functional machinery division, a total of fifty-four prizes will be distributed among ten sub-classifications, namely, metal cutting, metal forming, electrical, prime movers, conveying, pumps and compressors, jigs and fixtures, and miscellaneous parts.

In the division of industry machinery, prizes to the number of fifty-four will be divided among ten sub-classifications, namely process, construction, petroleum, steel making, farming, household, food-making, textile and clothing, printing, and activities not otherwise classified.

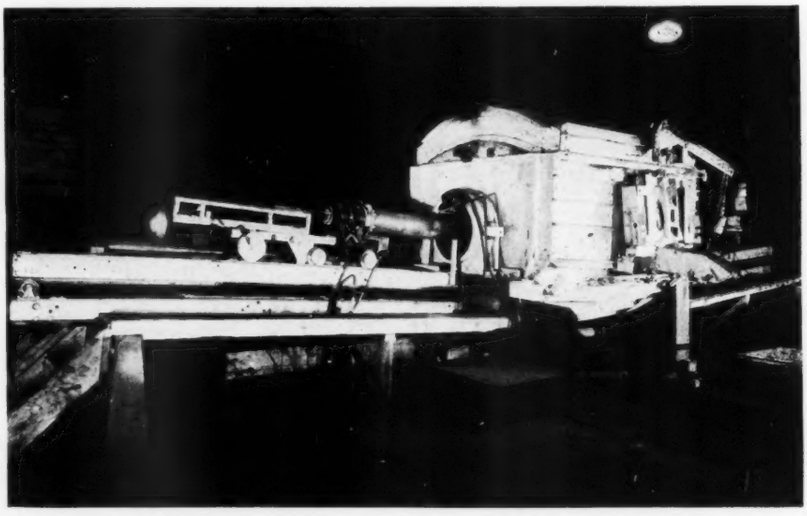
Submitted papers must describe either the redesign of an existing machine, structure or building, so that welding may be applied to its manufacture, or the paper may deal with a present design, either in whole or in part, of a machine, structure, or building not previously manufactured.

In certain classifications, however, slightly different eligibility requirements obtain. In the divisions of commercial welding and welderies, for example, owners and operators of functioning establishments may enter the competition with papers which describe details for successfully conducting such a business.

Contestants, it was announced, must have papers in duplicate on file with the Secretary of the Foundation, at Cleveland, Ohio, not later than June 1, 1938. Prospective entrants should communicate promptly with the Foundation Secretary, A. F. Davis, P.O. Box 5728, Cleveland, Ohio, for complete details of the rules and conditions covering this competition.

Ludlum Steel Puts New Furnace in Use

INSTALLATION of a hollow-electrode type furnace at the Ludlum Steel Company in Watervliet, New York, was completed recently and the furnace is now in full operation. Using the indirect arc, the furnace has



Electric furnace has two hollow electrodes through which briquetted refining reagents are pushed into arc

two rotating hollow electrodes 10 in. in diameter which are inclined slightly from the horizontal. A continuous looped chain equipped with lugs pushes refining reagents through 4½ in. holes in the electrodes. The velocity of the chain can be varied to obtain different rates of feed. The refining reagents are a mixture of suitable ore and pulverized coke made into briquettes which are baked to eliminate moisture. Smelted at very high temperatures at the arc, the metal is refined before it drops into the bath. An atmosphere of carbon monoxide is produced by the reactions.

The furnace tilts about the center of the electrodes which consequently do not have to be withdrawn when the furnace is tapped. Water-cooled electrode bushings are fastened to the heavy trunnions which support the furnace. Scrap and alloys to be melted and refined are charged on the hearth through a door similar to the method used in charging standard electric furnaces. Heats produced range from 1,000 to 6,000 lb. of molten metal. Power consumption averages approximately 550 kw. per hr. The current is single phase.

London Conference on Lubrication

THE Council of the Institution of Mechanical Engineers, England, is now organizing a "general discussion on lubrication and lubricants" to be held in London in the latter part of October. It is planned that this discussion will occupy a period of two days and will center around a series of about 100 papers which are being invited from leading authorities throughout the world. No individual paper may exceed 3,000 words while an average of 2,000 words is desired. Although manuscripts in foreign languages will be offered, all accepted papers will be printed in English as will also the reported discussions.

In their announcement of this conference the Council of the Institution of Mechanical Engineers calls attention to the fact that although considerable information can be found scattered in the technical literature, there is not available a comprehensive review of the state of current knowledge. It is however recognized that the fundamental characteristics of lubrication are imperfectly understood and the solution of many of the problems involved represents a field of effort in which all concerned can usefully contribute knowledge, experiences and views. The primary objects of the conference will be an endeavor to establish a correlation between theory and practice, to show how bearing design can be applied, to consider the relationship between static and kinetic friction, to balance academic research with engineering practice, to classify the views upon bearing metals and to review the significance of laboratory tests.

The American Society of Mechanical Engineers has accepted the invitation of the Council of the Institution of Mechanical Engineers to participate in this conference and it is expected that a considerable number of authorities on lubrication in this country will contribute papers.

Fibrous Glass Finds New Applications

SPEAKING before the Arkwright Club. New York City, Donald C. Simpson, expert from the fibrous glass laboratory of the Owens-Illinois Glass Company, explained some of the applications just being made of fibrous glass and their significance to industry generally.

To sound out the possibility in the uses of fibrous glass, the Owens-Illinois Glass Company established a separate research organization at its Evansville, Indiana plant. The first real development in fibrous glass was the creation of an air filter, consisting of fine glass fibers impregnated with an adhesive. Used in warm air heating systems, air conditioning units and in other manners, such filters have given remarkable results in cleaning air of pollen and other foreign matter.

Further developments led to a fibrous insulation glass with a density of only 1½ lb. per cu.ft., as compared to more than 5 lb. per cu.ft. for fibrous glass produced in the old way, and a density of 150 lb. per cu.ft. for solid glass. Because the fibrous insulation glass is only 1 per cent glass and 99 per cent air, a fact of vital importance in insulation, it is extremely lighter than the solid material.

For textile uses there has been evolved a glass fiber 15 in. long, and only 1/20 the diameter of a human hair. One pound of solid glass could be converted into such a fibrous thread nearly 5,000 mi. long.

There is now being offered fibrous glass insulation material in a variety of forms for ice refrigerators, for refrigerated railroad cars and similar applications. It is also being produced in the form of blankets for insulation of ovens, boilers and ducts, and for pipe insulation.

Mr. Simpson stated that intensive laboratory experiments are now under way in fabricating and weaving glass yarn or thread into various types of glass cloth for industrial purposes. As he pointed out, the present glass yarns do not have the softness and flexibility that it is hoped to ultimately obtain. The desired glass textile material is expected to have applications where a fire-proof and acid-proof material and resistance to rust and decay are of great importance.

MEETINGS

National Oil Burner and Air-Conditioning Exposition and Convention—Commercial Museum, Philadelphia, Pa. March 15–19. Oil Burner Institute, G. Harvey Porter, managing director, 30 Rockefeller Plaza, New York, N. Y.

Packaging Exposition—Pennsylvania Hotel, New York, N. Y. March 23–26. Roberts Everett Associates, show managers, 232 Madison Ave., New York, N. Y.

International Association for Testing Materials — International Congress, London, England, April 19–24. K. Headlam-Morley, 28 Victoria St., London, S.W.1.

American Foundrymen's Association — Annual convention and foundry show, Milwaukee, Wis., May 3–7. Author J. Tuscany, secretary, 632 Penton Building, Cleveland, Ohio.

Do You Know That—

• Stack cutting is used for flame cutting pieces of identical shape. The sheets to a total thickness of 3 in. or more are piled up and cut to the desired contour with an oxy-acetylene flame. (43)

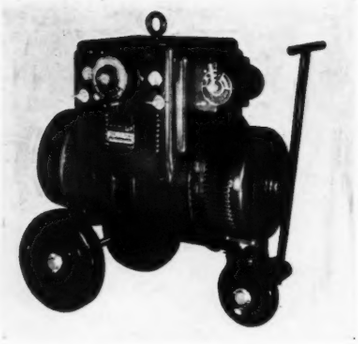
• Perfect circles can be made to look like spirals merely by shading the background in a pattern.

• Color alone may cause a variation of as much as 10 per cent in the estimation of distance. (44)

• Using a standard duplicating machine, the Lord's prayer as engraved on a flat surface, 2-in. in diameter, was duplicated within a circle, 0.005 in. diameter, which is equivalent to the diameter of the point of a rather dull pin. All of the 300 characters are plainly legible under the microscope. (45)

• In the new 12-car streamlined stainless steel trains built by the Edward G. Budd Manufacturing Company, 97 V-belts are used for driving auxiliary units such as engine fans, air-conditioning units, auxiliary generators, exciters and similar equipment. (46)

New Materials and Parts



Selective Control

Incorporated in arc welder for controlling motor horsepower. A latch locks the handle in the low-position, where only half the rated motor horsepower is used for starting and for welding up to ½ the rated generator capacity in continuous manual arc welding and up to 2/3 rated capacity for intermittent welding. For operation at higher rates a lever releases the handle for turning to the high-position where the full rated horsepower of the motor is available. Built in a series of seven ratings. Hobart Brothers, Troy, Ohio

Corrosion-Resistant Finish

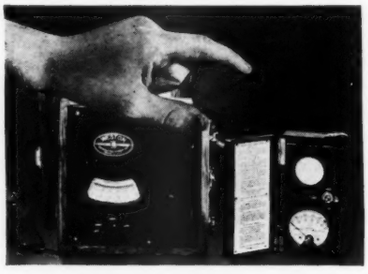
Suitable for materials that are subjected to forming operations because of its flexibility. Chemically resistant to corrosive and finish destroying agents like acids, alkalis, alcohols, disinfectants and cleaning solutions. Known as Roxyn C it is available in all colors of air-dry or bake-type finishes. Roxalin Flexible Lacquer Co., Inc., Elizabeth, N. J.

Flexible Coupling

Designed to eliminate lateral thrust of shaft. To provide for parallel misalignment four meshing gear-sector members fitted with pins turn on studs which are carried by one half of coupling. The pins on the gear sectors support rollers which fit into spherical pockets in the other half of coupling. The rollers permit angular misalignment. A dust and oil-tight cover is screwed on one flange to protect the working parts. Available in shaft sizes ¾ in. and larger, and in 3 to 500 hp. capacities. Harris Coupling Co., 265 W. 40th St., New York, N. Y.

Photo-cell Filter

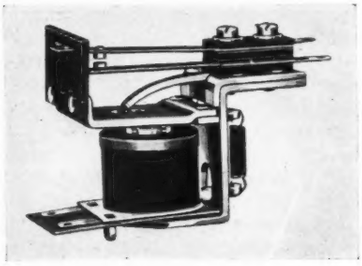
Complete cut-off at both ends of the visible spectrum brings the spectral sensitivity of the Photronic photoelectric cell into close agreement with that of the human eye. Recommended for critical photometric work particularly with monochromatic light. The filter is made of glass, and is intended for use with direct-reading photometric instruments in which a cell converts light energy to electrical energy that actuates an indicating meter. The total transmission of the Viscor filter is 37.5 per cent approximately. Weston Electrical Instrument Corp., Newark, N. J.



Mercury Switch

Housed in a metallic envelope, mercury to mercury make and break contact, is hydrogen filled and sealed against leakage. The point of circuit make and break is localized. Deionizing the arc promotes quick operation.

Designed for installation in special clips similar to cartridge clips. Operates by tilting through an angle of approximately 10 deg. from the horizontal. Reversing the switch in its clip reverses the make and break positions. Known as Staley Metallic Mercury Switch, can be supplied in 5, 10 and 15 amp. capacities in various types. Electric Switch Corp., Columbus, Ind.



A.C. Relay Unit

Has a floating armature designed to reduce friction, binding and sticking at the hinge-joint. A shading ring attached to the core is said to eliminate hum and chatter. Contact springs are phosphor bronze. Standard contact points are silver; other metals can be furnished to suit specifications. Contact combinations available range from single-pole single-throw to four-pole double-throw. All parts are cadmium plated. Relay is mounted by machine screw threaded into the bottom of the core. Draws approximately 2 watts. Supplied to operate on voltages up to 220 and on any standard cycle. Guardian Electric Mfg. Co., 1629 W. Walnut St., Chicago, Ill.

Control Thermostat

Stainless steel, corrosion-proof construction, for commercial service installations. Low-pressure and temperature controls normally are circuit-

closing at high pressures and temperatures. High rating is possible because of wide break when overload trips, heavy contact pressures and large contact surfaces. Range calibration indicator is visible through a window. Both range and differential are adjustable from the outside. Rated at: ¾ hp., 110 to 220 v., a.c.; and 1/3 hp., 115 v., d.c. Ranco, Inc., Columbus, Ohio.

Luminous Tube Transformers

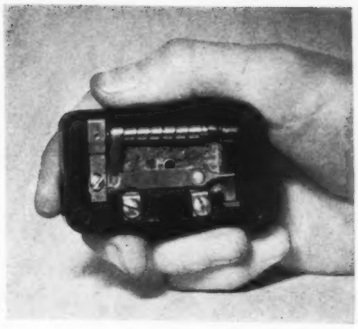
Designed to meet the requirements of the sign industry. Primary coils are wound with a paper insulated quadruple enameled wire which puts insulation between each turn. Secondary coils are wound with quadruple enameled wire with glassine paper between layers of wire. After assembling the coils and core in the case the unit is filled with an insulating compound. The cases are made of rust-resisting steel with welded joints and are given a black crinkle finish. Available for all standard secondary ratings, ranging from 2,000 v., 18 milliamperes up to 15,000 v., 60 milliamperes. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

D-H-S Bronzes

Available for castings or forgings, have a fine and dense grain structure. High compressive strength, ductility, hardness, resistance to acids, and ability to withstand wear and shock are properties claimed for these alloys. D-H-S bronzes are offered in four grades. Bartlett Hayward Div., Koppers Co., Baltimore, Md.

Thermal Overload Switch

Designed to be mounted on conduit or terminal box of motor. Automatically disconnects the motor from the line when an overload occurs before a dangerous temperature is reached. A



heater helix in series with the motor winding surrounds a bimetallic helix that is fixed at one end. The opposite end of the bimetallic helix carries a slotted arm which rotates about one support of a bowed snap spring which carries a contact arm. The Thermo-Tector is available for a.c. fractional hp. motors with 1.3 to 10.4 amp. rated full-load currents. General Electric Co., Schenectady, N. Y.

Dip-Spra Bonderizing

Uses a chemical to produce a corrosion inhibiting paint base coating on galvanized, zinc alloy and cadmium surfaces, as well as on iron and steel. The coating is integral with the metal and has a slight porosity that forms a base for any type of paint finish. Equipment required consists of a steam heated processing tank of suitable size in which a stainless steel spray pipe is placed above the solution level and a small circulation pump. Large or deep tanks are fitted with a spray pipe on each side. Parker Rust-Proof Co., Detroit, Mich.

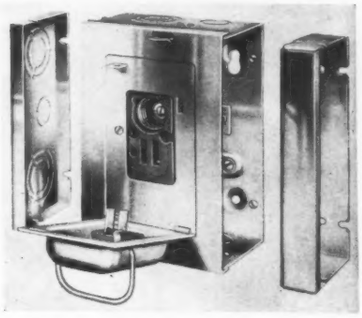
Rubber-Tired Wheel

Tire forms its own tube. Unit is assembled by putting the wheel inside of the tire. Wheel is double-disc type held together by axle sleeve and small bolts. Designed to be used on industrial trucks, carts, wheelbarrows, boat

dollies and portable display stands. Supports loads of 200 lb. per wheel at 12 to 15 lb. air pressure in tire. Tire size is 10x2¾ in. Wheel is also furnished with a caster swivel fork. Musselman Products Co., 6227 St. Clair Ave., Cleveland, Ohio.

Pullout Switch

Switching action is obtained by merely opening and closing the cover on the inside of which is mounted the switch blades. The main cover is a "dead front" plate that protects the



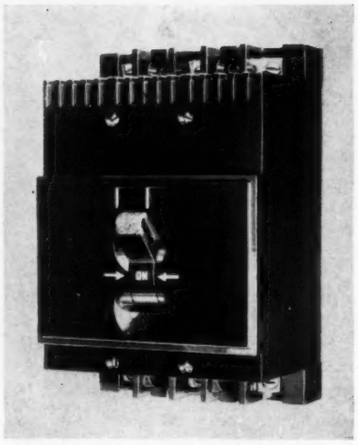
operator from accidental contact with live parts. Designed for close banking, endwalls are removable. By using special coupling straps, two or more switches can be connected to form a continuous trough for wiring. Colt's Patent Fire Arms Mfg. Co., Hartford, Conn.

Chroprime

Is a clear primer which can be applied to a chromium surface by spraying or dipping after which it is baked. It is used in the production of automobile name plates and for other purposes where colors are wanted on a corrosion-proof base. Provides a foundation for any kind of lacquer enamel. It is said to resist heat and to remain flexible. Metal may be engraved after primer is applied. Maas & Waldstein Co., 420 Lexington Ave., New York, N. Y.

Circuit Breaker

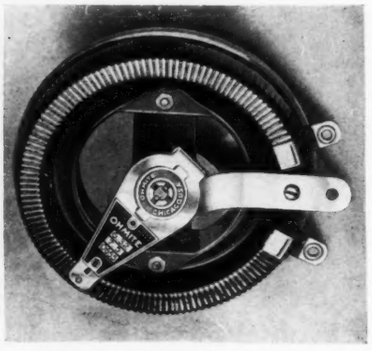
Automatic overload and short-circuit breaker action is controlled by a hermetically-sealed magnetic trip device. Can be reclosed as soon as current has returned to normal. Trip point is unaffected by ambient temperature. Time-delay units are offered that function on one of four time-overload curves. Units are available in plastic molded cases, steel safety cabinets and cast metal housings for hazardous locations, available in



ampere ratings as great as 35, at 250 v., d.c., or 460 v., a.c., with a choice of poles. Heinemann Electric Co., Trenton, N. J.

Power Rheostat

For controlling motor speeds, for lamp dimming devices, and for generator field control of welding machines. Constructed of metal and ceramic materials throughout. Resistance wire is wound over a horseshoe-shaped porce-



lain core. The spring-held contact arm is fitted with a universal-joint type contact shoe which spans several steps at all times. The rheostat is 12 in. in diam., and is rated at 1,000 watts. Ohmite Manufacturing Co., 4835 W. Flournoy St., Chicago, Ill.

Zin-O-Lyte

Process for plating brilliant deposits of bright zinc direct from the bath without bright dipping. Operates both in still and in barrel plating and is applicable to almost all types of iron and steel products. The deposits are ductile and firmly adherent and afford great rust protection. A process has been developed for the control of the corrosion of the zinc anode whereby the zinc content of the electrolyte is held within normal limits. E. I. du Pont de Nemours & Co., Wilmington, Del.

MANUFACTURERS' PUBLICATIONS

Acetylene—Air Reduction Sales Co., 60 East 42d St., New York, N. Y. Booklet, 12 pages, 3¾x9 in. Outlines the properties, uses and advantages of acetylene as a fuel gas for welding and cutting.

Arc Welding—Hobart Brothers Co., Troy, Ohio. Catalog, 8½x11 in., 23 pages. Illustrating features, design and construction of motor and gasoline engine driven arc welders with accessories and field of applications.

Bearings — SKF Industries, Inc., Front St. & Erie Ave., Philadelphia, Pa. 23 pages, 8½x11 in. Illustrating with photographs and line drawings the applications of ball and roller bearings in equipment for the oil industry. Tables on the load-carrying capacity of spherical roller bearings are included.

Beryllium Copper—Beryllium Corporation of Pennsylvania, Reading, Pa. Folder 4x9 in. Describes properties, heat-treatment and uses of beryllium copper.

Chains and Sprockets for Power Transmission — Baldwin-Duckworth Chain Corp. Springfield, Mass. Catalog L, 72 pages, 8½x11 in. Presents engineering data, specifications and uses for roller, cable or leaf, block and special chains, also sprockets and flexible couplings.

Dairy Temperature Recorder—Consolidated Ashcroft Hancock Co., Inc., Bridgeport, Conn. Bulletin, 8 pages, 8½x11 in. Illustrates temperature recorder with types of charts available and interchangeable fittings, also pasteurizing thermometers.

Developments in Drop Forging—Chambersburg Engineering Co., Chambersburg, Pa. Bulletin, 13 pages, 8½x11 in. A treatise outlining recent developments in modern drop forging practice.

Dip-Spra Bonderizing—Parker Rust-Proof Co., Detroit, Mich. Folder, 8½x11 in. Describes a method of applying a corrosion resisting paint base on iron, steel, galvanized, zinc alloys or cadmium surfaces.

Electric Heating Elements—Harold E. Trent Co., 618-640 N. 54th St., Philadelphia, Pa. Leaflets, TC 30—revised, TD-11, and TB-26, 8x11 in. Describes types, applications, and construction details of electric heating units. Many illustrations are shown including dimensional cuts and tables along with electrical characteristics.

Electronic Tubes — Westinghouse Lamp Co., Bloomfield, N. J. Bulletin No. 8, 4 pages, 8½x11 in. Describes uses, construction and characteristics of vacuum type and gas-filled type phototubes.

Enamels, Equipment and Supplies—Ferro Enamel Corp., Cleveland, Ohio. Catalog "B" 41 pages, 8½x11 in. Illustrates and describes typical installations of porcelain enameling furnaces and complete enameling plants as well as available equipment and supplies.

Explosion Tested D.C. Motors—Westinghouse Electric and Manufacturing Co., East Pittsburgh, Pa. Leaflet 20652. Describes the improved design of Type SK d.c. motors, for use in withstanding internal explosions of gas or coal dust without injuring the equipment and without emitting flames.

Flexible Couplings—Boston Gear Works, Inc., North Quincy, Mass. Folder, 3½x6 in. Describes special ball bearing type flexible couplings.

Flexible-Shafts and Equipments—Swartz & White Mfg. Co., Binghamton, N. Y. Catalog, 32 pages, 4x9¼ in. Describes selection, uses and care of flexible shaft drives and applications to small tools. Also illustrates and lists various attachments for shop use.

Flocontrol Valves — Consolidated Ashcroft Hancock Co., Inc., Bridgeport, Conn. Bulletin, 4 pages, 7¾x10¾ in. Describes a line of screw stem valves with micrometer dials and pointers which indicate valve opening.

Forgings—The Champion Machine & Forging Co., Cleveland, Ohio. Folder, 8 pages, 8½x11 in. Showing some of the kinds of work this company can do; also describes and illustrates plant and testing equipment.

Galvanized Paintgrip Sheets—American Rolling Mill Co., Middletown, Ohio. Folder, 8½x11 in. Describes a galvanized sheet metal that is paintable without special acid treatments.

Industrial Instruments and Scientific Apparatus—James G. Biddle Co., 1211-13 Arch St., Philadelphia, Pa. Bulletin 1456. Eight pages, 7¾x10½ in. Illustrates with brief descriptions a few important specialties for testing insulation, grounds and circuits, also includes information on vibrating-reed tachometers and frequency meters.

Kanthal Alloys for High Temperatures—The C. O. Jelliff Mfg. Co., Southport, Conn. Catalog No. 34, 56 pages. 6x9 in. Outlines the mechanical and chemical properties of Kanthal alloy materials used for electric heating apparatus of all kinds. Commercial sizes of wire and ribbon are listed with current temperature characteristics, surface radiating factors, and weights.

Materials Handling Machinery—Gifford-Wood Co., Hudson, N. Y. Catalog 136, 112 pages, 7½x10½ in. Contains detailed descriptions of equipment, with many tables, line cuts, and illustrations of improved designs and construction of machinery for materials handling systems for nearly every phase of industrial activity.

Non-Ferrous Nickel Alloys—The International Nickel Co., Inc., 67 Wall St., New York, N. Y. Booklet "Monel for Mechanical Jobs," 48 pages, 9x12 in. Guide book covering corrosion resistance and other properties of Monel

and other non-ferrous nickel alloys. Also describes some of the newer forms of Monel, including "K" Monel, "S" Monel, etc. Contains 20 subdivisions, well illustrated, devoted to specific problems in various fields.

Pontalite—E. I. duPont de Nemours & Co., Inc., 350 Fifth Ave., New York, N. Y. Booklet, 18 pages, 3½x6¼ in. Covers physical, chemical, thermal, optical, electrical and working properties of cast product and the molding powder Pontalite. Contains notes on compression and injection molding.

Roller Chain Flexible Coupling—Whitney Chain & Mfg. Co., Hartford, Conn., Bulletin V-110A, 8 pages, 8½ x11 in. Advantages, construction, applications and sizes with service factors and horsepower ratings of new line of couplings.

Rol-Man Manganese Steel—Manganese Steel Forge Co., Philadelphia, Pa., Bulletin R-56, 28 pages, 3½x6¼ in. An index of applications of rolled manganese steel for mechanical purposes, its use in the form of welding rod and threaded bolts.

Rotary Converters—Janette Mfg. Co., 556 West Monroe St., Chicago, Ill. Bulletin 13-1, folder. Types of machines used to convert d.c. to a.c. for operating small apparatus and instruments.

Rotary Vacuum Pumps—Beach-Russ Co., 50 Church St., New York, N. Y. Bulletin No. 70, Folder 8½x11 in. Construction details, operating principles, types and sizes of rotary vacuum pumps for dry or semi-wet work.

Shield-Arc SAE Welder—The Lincoln Electric Co., Cleveland, Ohio. Bulletin No. 316, 4 pages, 8½x11 in. Specifications and description of dual continuous control shield-arc welders with dimension table.

Speed Control at Work—Reeves Pulley Co., Columbus, Ohio. Booklet, 32 pages, 4½x6 in. Description and illustrations of the methods employed by twelve well-known manufacturers to secure speed adjustments on production machines.

Temperature Relays—General Electric Co., Schenectady, N. Y. Bulletin GEA-2525, 8x10½ in. 4 pages. Type ICT-self resetting temperature relays are described for protecting a.c. machines and transformers against abnormal heating.

V-Belt Drives—Rockwood Mfg. Co. Indianapolis, Ind. Catalogs No. 782 and 783, 12 and 32 pages respectively, 8¾x11½ in. Data books covering V-belt drives planned to help the engineer in making the proper selection of drives.

Vibro-Insulators—B. F. Goodrich Co., Akron, Ohio. Catalog 7900, 4 pages, 8½x11 in. Contains specifications, deflection tables and technical information concerning Vibro-Insulators.

Zinc Base Die-Casting Alloys—Apex Smelting Co., 2554 Fillmore St., Chicago, Ill. Metalgram Folders No. 17 and 18. 8½x11 in. Graphs showing tensile strength and stability of No. 6 zinc base alloy.

BOOKS AND BULLETINS

A.S.T.M. Standards —1936
Part I. Metals, Part II. Non-Metallic Materials

American Society for Testing Materials, 6x9¼ in. Part I. 898 pages, Part II, 1,477 pages. Published by American Society for Testing Materials, 260 S. Broad St., Philadelphia, Pa. Prices: for either part separately, in cloth $7.50, in half leather $8.50; for both parts, in cloth $14, in half leather $16.

This triennial publication contains all of the standard specifications, methods of test, recommended practices and definitions formally adopted by the Society. In each Part, the specifications for a partcular class of material are given first, followed directly by the test methods and definitions.

Of the 181 standards in Part I, 109 cover the ferrous metals, while 60 relate to non-ferrous metals. Twelve of the standards involve metallography and general testing methods.

Part II contains many new standards and revisions which were adopted during 1936 on non-metallic materials.

Flow Meter Engineering Handbook

Louis Gess and *J. O. Feldmark.* 162 pages, 8x10½ in. 28 illustrations, 36 tables. Grain leather covers. Published by The Brown Instrument Co., Wayne & Roberts Aves., Philadelphia, Pa. Price $2.

The purpose of this handbook is to present the principles underlying the design, installation and operation of flow meters. An outline is given of the essential features common to all types of fluid measuring equipment using an orifice. The book is divided into eight parts, covering general principles, orifice design and flow measurement of steam, water, liquid, air and gas. Charts and photographs are used to illustrate the text. Typical power and processing problems are used to show applications of formulas.

Aircraft Engines

Arthur B. Domonoske and Volney C. Finch. 342 pages, 6x9¼ in., numerous graphs and tables. Clothboard covers. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York, N. Y. Price $3.75.

Basic theory and principles of aircraft engine design and operation are presented in a manner intended to stimulate and to develop a spirit of investigation and self-reliance. The further promotion of progress relative to aircraft power plants is emphasized as desirable.

A short introduction covering types of aircraft engines, trend of design and characteristics is followed by a chapter devoted to the thermodynamic theory of heat and gases. Internal combustion engine cycles and ideal engine efficiencies are described. Constitution of petroleum fuels, the chemistry of combustion, detonation and antidetonants and factors which affect combustion in the aircraft engines are fully covered. Thermal efficiency, supercharging, waste heat, cooling and testing are other subjects treated. Each chapter has references and problems.

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Plastic Flow of Metals

Earle B. Norris. 30 pages, 6x9 in. 19 diagrams, 3 tables. Paper covers. Engineering Experiment Station Series, Bulletin No. 27. Published by The Virginia Polytechnic Institute, Blacksburg, Va. Price 20 cents.

Observations are cited to indicate that the plastic flow of metals and alloys under tension, throughout the range from yield point up to rupture, follows one continuous law with constants to be determined for each metal. A method for evaluating the flow constants is presented. The thesis is stated following a thorough discussion of stress-strain diagrams, cold-working in tension, curves of true stress verus elongation, reduction of diameter and true elongation, also log true stress vs. log true elongation.

Tin Research

Recent reports, published by the International Tin Research Development Council. Copies are available free from L. J. Tavener, 149 Broadway, New York, N. Y.

"Micro-Plasticity in Crystals of Tin", Series A, No. 44, 20 pages.

"The Retarding Effect of Stannous Salts on the Oxidation of Olein and Oils," Series A, No. 45, 10 pages.

"The Spectrographic Analysis of Tin," Series A, No. 46, 26 pages.

Lessons in Arc Welding

The Lincoln Electric Co., Cleveland, Ohio. Loose-leaf mimeographed. 130 pages, 8x11 in. Price 50 cents.

A new edition in considerably enlarged form that provides working information covering the application and use of arc welding. The volume contains 44 lessons which are based on a course conducted by the company in its plant welding school. Beginning with general fundamental suggestions, the lessons take the student by easy stages through all the important phases of the subject. Many sketches are used to illustrate and make clear the text.

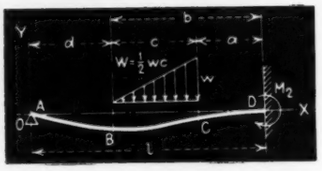
Formulas For Beams

II—STATICALLY INDETERMINATE CASES

(*Continued from page 80, February number*)

Loading, Support and Vertical Shear V	Reactions R_1 and R_2 and Constraining Moments M_1 and M_2, Bending Moment M, Max. Positive and Negative Bending Moments, Deflection y, Max. Deflection and End Slope θ

26—ONE END FIXED, ONE END SUPPORTED. PARTIAL TRIANGULAR LOAD



$$R_1 = \frac{1}{20}\frac{W}{l^3}\left[(10\,ab+15a^2+5b^2)\,l-4a^3-2ab^2-3a^2b-b^3\right] \quad ; \quad R_2 = W - R_1$$

$$M_2 = R_1 l - \frac{1}{3}\,W\,(2\,a+b) \quad ; \quad \text{(A to B)},\ M = R_1 x$$

$$\text{(B to C)},\ M = R_1 x - \frac{1}{3}\,W\,\frac{(x-d)^3}{c^2} \quad ; \quad \text{(C to D)},\ M = R_1 x - W\left(x - d - \frac{2}{3}c\right)$$

$$\text{Max.} + M = R_1\left(d+\frac{2}{3}\,c\,\sqrt{\frac{R_1}{W}}\right),\ \text{at } x = d + c\,\sqrt{\frac{R_1}{W}} \quad ; \quad \text{Max.} - M = -M_2$$

$$\text{(A to B)},\ y = \frac{1}{EI}\left[R_1\left(\frac{1}{6}x^3-\frac{1}{2}l^2x\right)+Wx\left(\frac{1}{2}a^2+\frac{1}{3}ac+\frac{1}{12}c^2\right)\right]$$

$$\text{(B to C)},\ y = \frac{1}{EI}\left[R_1\left(\frac{1}{6}x^3-\frac{1}{2}l^2x\right)+Wx\left(\frac{1}{2}a^2+\frac{1}{3}ac+\frac{1}{12}c^2\right)-W\,\frac{(x-d)^5}{60c^2}\right]$$

$$\text{(C to D)},\ y = \frac{1}{EI}\left\{R_1\left(\frac{1}{6}x^3-\frac{1}{2}l^2x+\frac{1}{3}l^3\right)+W\left[\frac{1}{2}\left(a+\frac{1}{3}c\right)^2x-\frac{1}{6}\left(x-d-\frac{2}{3}c\right)^3+\frac{1}{6}\left(a+\frac{1}{3}c\right)^3-\frac{1}{2}\left(a+\frac{1}{3}c\right)^2 l\right]\right\}$$

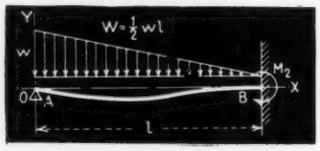
$$\text{(A to B)},\ V = R_1$$

$$\text{(B to C)},\ V = R_1 - W\left(\frac{x-d}{c}\right)^2$$

$$\text{(C to D)},\ V = R_1 - W$$

$$\theta = -\frac{1}{EI}\left[\frac{1}{2}R_1l^2 - W\left(\frac{1}{12}c^2+\frac{1}{3}ac+\frac{1}{2}a^2\right)\right],\ \text{at A}$$

27—ONE END FIXED, ONE END SUPPORTED. TRIANGULAR LOAD



$$R_1 = \frac{11}{20}\,W \quad ; \quad R_2 = \frac{9}{20}\,W \quad ; \quad M_2 = \frac{7}{60}\,Wl \quad ; \quad M = W\left(\frac{11}{20}\,x - \frac{x^2}{l} + \frac{1}{3}\,\frac{x^3}{l^2}\right)$$

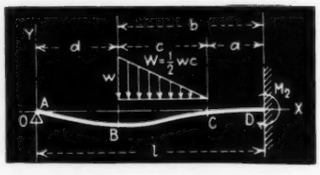
$$\text{Max.} + M = 0.0846\,Wl,\ \text{at } x = 0.329l \quad ; \quad \text{Max.} - M = -\frac{7}{60}\,Wl,\ \text{at B}$$

$$y = \frac{1}{120}\,\frac{W}{EIl}\left(11\,lx^3 - 3l^3x - 10\,x^4 + 2\,\frac{x^5}{l}\right) \quad ; \quad \text{Max. } y = -\,0.00609\,\frac{Wl^3}{EI},\ \text{at } x = 0.402l$$

$$V = W\left(\frac{11}{20} - \frac{2x}{l} + \frac{x^2}{l^2}\right)$$

$$\theta = -\frac{1}{40}\,\frac{Wl^2}{EI}\ \text{at A}$$

28—ONE END FIXED, ONE END SUPPORTED. PARTIAL TRIANGULAR LOAD



$$R_1 = \frac{1}{20}\,\frac{W}{l^3}\left[(10ab+5a^2+15b^2)\,l-a^3-2a^2b-3ab^2-4b^3\right] \quad ; \quad R_2 = W - R_1$$

$$M_2 = R_1 l - \frac{1}{3}\,W\,(a+2b) \quad ; \quad \text{(A to B)},\ M = R_1 x$$

$$\text{(B to C)},\ M = R_1 x - \frac{(x-d)^2}{c}\,W + \frac{(x-d)^3}{3c^2} \quad ; \quad \text{(C to D)},\ M = R_1 x - W\left(x-d-\frac{1}{3}\,c\right)$$

$$\text{Max.} + M = R_1\left(d+c-c\,\sqrt{1-R_1/W}\right) - \frac{1}{3}\,Wc\left(1-\sqrt{1-R_1/W}\right)^2\left(2+\sqrt{1-R_1/W}\right),$$

$$\text{at } x = d+c\left(1-\sqrt{1-R_1/W}\right)$$

$$\text{Max.} - M = -M_2,\ \text{at D}$$

$$\text{(A to B)},\ y = \frac{1}{EI}\left\{R_1\left(\frac{1}{6}x^3-\frac{1}{2}l^2x\right)+Wx\left(\frac{1}{2}a^2+\frac{2}{3}ac+\frac{1}{4}c^2\right)\right\}$$

$$\text{(B to C)},\ y = \frac{1}{EI}\left\{R_1\left(\frac{1}{6}x^3-\frac{1}{2}l^2x\right)+W\left[\frac{1}{60}\,\frac{(x-d)^5}{c^2}-\frac{1}{12}\,\frac{(x-d)^4}{c}+\left(\frac{1}{2}a^2+\frac{2}{3}ac+\frac{1}{4}c^2\right)x\right]\right\}$$

$$\text{(C to D)},\ y = \frac{1}{EI}\left\{R_1\left(\frac{1}{6}x^3-\frac{1}{2}l^2x+\frac{1}{3}l^3\right)-\frac{1}{2}W\left(a+\frac{2}{3}c\right)^2(l-x)\right\}$$

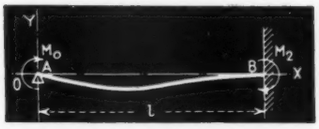
$$\text{(A to B)},\ V = R_1$$

$$\text{(B to C)},\quad V = R_1 - \frac{2\,(x-d)}{c}\,W + \frac{(x-d)^2}{c^2}\,W$$

$$\text{(C to D)},\ V = R_1 - W$$

$$\theta = -\frac{1}{EI}\left[\frac{1}{2}R_1l^2 - W\left(\frac{1}{2}a^2+\frac{2}{3}ac+\frac{1}{4}c^2\right)\right],\ \text{at A}$$

29—ONE END FIXED, ONE END SUPPORTED. END COUPLE



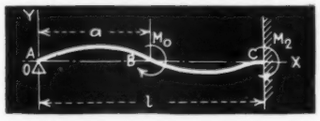
$$R_1 = -\frac{3}{2}\frac{M_0}{l} \quad ; \quad R_2 = +\frac{3}{2}\frac{M_0}{l} \quad ; \quad M_2 = \frac{1}{2}M_0 \quad ; \quad M = \frac{1}{2}M_0\left(2-3\frac{x}{l}\right)$$

Max. $+M = M_0$, at A ; Max. $-M = -\frac{1}{2}M_0$, at B ; $y = \frac{1}{4}\frac{M_0}{EI}\left(2x^2 - \frac{x^3}{l} - lx\right)$

$$V = -\frac{3}{2}\frac{M_0}{l}$$

Max. $y = -\frac{1}{27}\frac{M_0 l^2}{EI}$, at $x = \frac{1}{3}l$; $\theta = -\frac{1}{4}\frac{M_0 l}{EI}$, at A

30—ONE END FIXED, ONE END SUPPORTED. INTERMEDIATE COUPLE



$$R_1 = -\frac{3}{2}\frac{M_0}{l}\left(\frac{l^2-a^2}{l^2}\right); \; R_2 = +\frac{3}{2}\frac{M_0}{l}\left(\frac{l^2-a^2}{l^2}\right) \quad ; \quad M_2 = \frac{1}{2}M_0\left(1-3\frac{a^2}{l^2}\right)$$

(A to B), $M = R_1 x$; (B to C), $M = R_1 x + M_0$

Max. $+M = M_0\left[1 - \frac{3a(l^2-a^2)}{2l^3}\right]$, at B (to right)

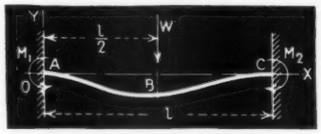
(A to B), $V = R_1$

(B to C), $V = R_1$

Max. $-M = -M_2$, at C, when $a < 0.275l$

Max. $-M = R_1 a$, at B, (to left), when $a > 0.275l$

31—BOTH ENDS FIXED. CENTER LOAD



$$R_1 = \frac{1}{2}W \quad ; \quad R_2 = \frac{1}{2}W \quad ; \quad M_1 = \frac{1}{8}Wl \quad ; \quad M_2 = \frac{1}{8}Wl$$

(A to B), $M = \frac{1}{8}W(4x-l)$; (B to C), $M = \frac{1}{8}W(3l-4x)$

(A to B), $V = +\frac{1}{2}W$

Max. $+M = \frac{1}{8}Wl$, at B ; Max. $-M = -\frac{1}{8}Wl$, at A and C

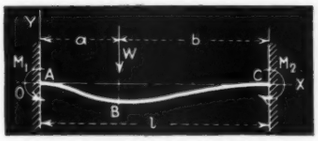
(B to C), $V = -\frac{1}{2}W$

(A to B), $y = -\frac{1}{48}\frac{W}{EI}(3lx^2 - 4x^3)$; Max. $y = -\frac{1}{192}\frac{Wl^3}{EI}$, at B

(A to B), $y = \frac{M_0}{EI}\left[\frac{l^2-a^2}{4l^3}(3l^2x - x^3) - (l-a)x\right]$

(B to C), $y = \frac{M_0}{EI}\left[\frac{l^2-a^2}{4l^3}(3l^2x - x^3) - lx + \frac{1}{2}(x^2+a^2)\right]$; $\theta = \frac{M_0}{EI}\left(a - \frac{1}{4}l - \frac{3}{4}\frac{a^2}{l}\right)$, at A

32—BOTH ENDS FIXED. INTERMEDIATE LOAD



$$R_1 = \frac{Wb^2}{l^3}(3a+b) \quad ; \quad R_2 = \frac{Wa^2}{l^3}(3b+a) \quad ; \quad M_1 = W\frac{ab^2}{l^2} \quad ; \quad M_2 = W\frac{a^2 b}{l^2}$$

(A to B), $M = -W\frac{ab^2}{l^2} + R_1 x$; (B to C), $M = -W\frac{ab^2}{l^2} + R_1 x - W(x-a)$

(A to B), $V = R_1$

(B to C), $V = R_1 - W$

Max. $+M = -W\frac{ab^2}{l^2} + R_1 a$, at B ; max. possible value $= \frac{1}{8}Wl$ when $a = \frac{1}{2}l$

Max. $-M = M_1$ when $a < b$; max. possible value $= -0.1481\,Wl$ when $a = \frac{1}{3}l$

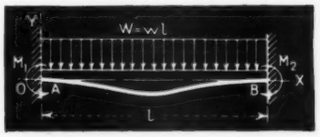
Max. $-M = -M_2$ when $a > b$; max. possible value $= -0.1481\,Wl$ when $a = \frac{2}{3}l$

(A to B), $y = \frac{1}{6}\frac{Wb^2x^2}{EIl^3}(3ax + bx - 3al)$

(B to C), $y = \frac{1}{6}\frac{Wa^2(l-x)^2}{EIl^3}\left[(3b+a)(l-x) - 3bl\right]$

Max. $y = -\frac{2}{3}\frac{W}{EI}\frac{a^3b^2}{(3a+b)^2}$, at $x = \frac{2al}{3a+b}$, if $a > b$; Max. $y = -\frac{2}{3}\frac{W}{EI}\frac{a^2b^3}{(3b+a)^2}$, at $x = l - \frac{2bl}{3b+a}$, if $a < b$

33—BOTH ENDS FIXED. UNIFORM LOAD



$$R_1 = \frac{1}{2}W \quad ; \quad R_2 = \frac{1}{2}W \quad ; \quad M_1 = \frac{1}{12}Wl \quad : \quad M_2 = \frac{1}{12}Wl$$

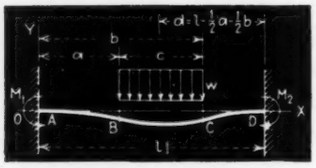
$$M = \frac{1}{2}W\left(x - \frac{x^2}{l} - \frac{1}{6}l\right); \quad \text{Max.} + M = \frac{1}{24}Wl, \text{ at } x = \frac{l}{2}$$

Max. $-M = -\frac{1}{12}Wl$, at A and B

$$V = \frac{1}{2}W\left(1 - \frac{2x}{l}\right)$$

$$y = \frac{1}{24}\frac{Wx^2}{EIl}(2lx - l^2 - x^2) \quad ; \quad \text{Max } y = -\frac{1}{384}\frac{Wl^3}{EI}, \text{ at } x = \frac{l}{2}$$

34—BOTH ENDS FIXED. PARTIAL UNIFORM LOAD



(A to B), $V = R_1$

(B to C), $V = R_1 - W\,\frac{x-a}{c}$

(C to D), $V = R_1 - W$

$$R_1 = \frac{1}{4}\,\frac{W}{l^2}\left(12\,d^2 - 8\,\frac{d^3}{l} + 2\,\frac{b\,c^2}{l} - \frac{c^3}{l} - c^2\right) \quad ; \quad R_2 = W - R_1$$

$$M_1 = -\,\frac{1}{24}\,\frac{W}{l}\left(24\,\frac{d^3}{l} - 6\,\frac{bc^2}{l} + 3\,\frac{c^3}{l} + 4\,c^2 - 24\,d^2\right)$$

$$M_2 = \frac{1}{24}\,\frac{W}{l}\left(24\,\frac{d^3}{l} - 6\,\frac{bc^2}{l} + 3\,\frac{c^3}{l} + 2c^2 - 48d^2 + 24dl\right)$$

(A to B), $M = -M_1 + R_1x$; (B to C), $M = -M_1 + R_1x - \frac{1}{2}\,W\,\frac{(x-a)^2}{c}$

(C to D), $M = -M_1 + R_1x - W\,(x - l + d)$

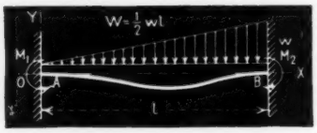
Max. $+M$ is between B and C at $x = a + \frac{R_1}{W}\,c$

Max. $-M = -M_1$ when $a < l - b$; Max. $-M = -M_2$ when $a > l - b$

(A to B), $y = \frac{1}{6EI}\,(R_1x^3 - 3\,M_1x^2)$; (B to C), $y = \frac{1}{6EI}\left(R_1x^3 - 3\,M_1x^2 - \frac{1}{4}\,W\,\frac{(x-a)^4}{c}\right)$

(C to D), $y = \frac{1}{6EI}\left[R_2\,(l-x)^3 - 3\,M_2\,(l-x)^2\right]$

35—BOTH ENDS FIXED. TRIANGULAR LOAD



$V = W\left(\frac{3}{10} - \frac{x^2}{l^2}\right)$

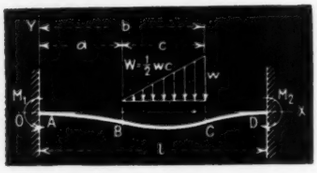
$$R_1 = \frac{3}{10}\,W \quad ; \quad R_2 = \frac{7}{10}\,W \quad ; \quad M_1 = \frac{1}{15}\,Wl \quad ; \quad M_2 = \frac{1}{10}\,Wl$$

$M = W\left(\frac{3}{10}\,x - \frac{1}{15}\,l - \frac{1}{3}\,\frac{x^3}{l^2}\right)$; Max. $+M = 0.043\,Wl$, at $x = 0.548l$

Max. $-M = -\frac{1}{10}\,Wl$, at B

$$y = \frac{1}{60}\,\frac{W}{EI}\left(3\,x^3 - 2\,lx^2 - \frac{x^5}{l^2}\right) \quad ; \quad \text{Max. } y = -\,0.002617\,\frac{Wl^3}{EI}, \text{ at } x = 0.525l$$

36—BOTH ENDS FIXED. PARTIAL TRIANGULAR LOAD



(A to B), $V = R_1$

(B to C), $V = R_1 - W\,\frac{(x-a)^2}{c^2}$

(C to D), $V = R_1 - W$

$$R_1 = \frac{W}{l^2}\left(3d^2 - \frac{1}{6}\,c^2 + \frac{1}{3}\,\frac{bc^2}{l} - \frac{17}{135}\,\frac{c^3}{l} - 2\,\frac{d^3}{l}\right) \quad ; \quad R_2 = W - R_1$$

$$M_1 = -\,\frac{W}{l}\left(\frac{d^3}{l} + \frac{1}{9}\,c^2 + \frac{51}{810}\,\frac{c^3}{l} - \frac{1}{6}\,\frac{c^2b}{l} - d^2\right)$$

$$M_2 = \frac{W}{l}\left(\frac{d^3}{l} + \frac{1}{18}\,c^2 + \frac{51}{810}\,\frac{c^3}{l} - \frac{1}{6}\,\frac{c^2b}{l} - 2d^2 + dl\right)$$

(A to B), $M = -M_1 + R_1x$; (B to C), $M = -M_1 + R_1x - W\,\frac{(x-a)^3}{2c^2}$

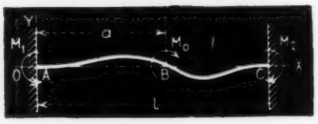
(C to D), $M = -M_1 + R_1x - W\,(x - l + d)$; Max. $+M$ at $x = a + c\sqrt{\frac{R_1}{W}}$

Max. $-M = M_1$ when $d > \frac{l}{2}$, $-M_2$ when $d < \frac{l}{2}$

(A to B), $y = \frac{1}{6EI}\,(R_1x^3 - 3\,M_1x^2)$; (B to C), $y = \frac{1}{EI}\left(\frac{1}{6}\,R_1x^3 - \frac{1}{2}\,M_1x^2 - \frac{1}{60}\,W\,\frac{(x-a)^5}{c^2}\right)$

(C to D), $y = \frac{1}{6EI}\left\{R_1\,(x^3 - 3l^2x + 2l^3) - 3\,M_1\,(l-x)^2 + W\left[3d^2x + d^3 - 3d^2l - (x - l + d)^3\right]\right\}$

37—BOTH ENDS FIXED. INTERMEDIATE COUPLE



$V = R_1$

$$R_1 = -\,6\,\frac{M_0}{l^3}\,(al - a^2) \quad ; \quad R_2 = 6\,\frac{M_0}{l^3}\,(al - a^2)$$

$$M_1 = \frac{M_0}{l^2}\,(4la - 3a^2 - l^2) \quad ; \quad M_2 = \frac{M_0}{l^2}\,(2la - 3a^2)$$

(A to B), $M = -M_1 + R_1x$; (B to C), $M = -M_1 + R_1x + M_0$

Max. $+M = M_0\left(4\,\frac{a}{l} - 9\,\frac{a^2}{l^2} + 6\,\frac{a^3}{l^3}\right)$, just to right of B

Max. $-M = M_0\left(4\,\frac{a}{l} - 9\,\frac{a^2}{l^2} + 6\,\frac{a^3}{l^3} - 1\right)$ just to left of B, if $a < 0.84l$

Max. $-M = -M_2$ if $a > 0.84l$

(A to B), $y = \frac{1}{6EI}\,(3\,M_1x^2 - R_1x^3)$

(B to C), $y = \frac{1}{6EI}\left[(M_1 + M_0)\,(3\,x^2 - 6\,lx + 3\,l^2) + R_1\,(3\,l^2x - x^3 - 2\,l^3)\right]$

Max. $+y$ at $x = \frac{2M_1}{R_1}$; Max. $-y$ at $x = l - \frac{2M_2}{R_2}$